REPORT OF BOARD ON

COMPARATIVE TRIALS OF THE SCOUT CRUISERS

Birmingham–Salem–Chester

December 22, 1909

GOVERNMENT PRINTING OFFICE
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CONTENTS.

COMPARATIVE TRIALS OF SCOUT CRUISERS.

BIRMINGHAM—SALEM—CHESTER.

NAVY DEPARTMENT,
BUREAU OF STEAM ENGINEERING,
Washington, D. C., December 22, 1909.

SIR: The Board appointed by the Navy Department, under orders dated October 21, 1908, to carry out comparative tests of the main and auxiliary machinery of the scout cruisers U. S. S. *Birmingham*, *Salem*, and *Chester*, having finished the work assigned to it, respectfully submits, as directed, the following report, embracing an analysis of the results obtained, together with a copy of the data recorded during the various tests made.

OBJECT AND DESCRIPTION OF TESTS.

The object of the tests was to ascertain and compare the steam economy of the three types of propelling machinery fitted in the vessels named, the hulls being practically of the same model, at various speeds while underway and under as nearly as possible identical conditions. To accomplish this, special water-measuring apparatus was built and installed in each ship for the tests, and so arranged that the weight of steam used by the propelling machinery, as well as the auxiliaries, could be, after condensation, separately and accurately ascertained. In order to make the comparison complete, six distinct series of tests were carried out, as follows:

I. Tests of auxiliary machinery (vessel in port).

II. Boiler evaporative tests (vessel in port).

III. Standardization runs. Each vessel was standardized over the Rockland measured mile course just prior to undertaking the (IV) steam-consumption tests. As will appear later, this involved standardizing the *Salem* on two different occasions, as two series of steam-consumption tests were carried out. In these trials, moreover, and as well as all other (V and VI) sea runs, the aim was to load each vessel so that the displacement for each trial should be as nearly as could be calculated in the beginning, an average of 4,000 tons. All ships were dry-docked prior to standardization and coal-consumption trials, and the bottoms cleaned and painted.

IV. Steam-consumption tests of main and auxiliary machinery (vessel under way).

V. Coal-consumption tests of about 1,000, 750, and 2,000 knots at speeds, respectively, of 10, 15, and 20 knots per hour.

VI. A full power run of twenty-four hours' duration. The *Birmingham* was unable to complete this trial on account of a defect which developed in the machinery. This vessel, after finishing twelve hours of the test, slowed down and returned to port.

It will be convenient to describe the different sets of tests in the order above mentioned, although they were not carried out in the sequence outlined.

As above stated, the scheme of comparison adopted was to ascertain by measurement of the exhaust steam the economy of not only the propelling machinery under varying conditions of sea speed, but in addition, of the various auxiliaries. It may be pointed out here that coal-consumption trials, even with the same quality of coal, weather, and condition of bottoms, do not furnish the means of accurate comparison, because of the personal element in firing, which can not be eliminated or satisfactorily allowed for. Moreover, the boiler plants on the *Birmingham* and *Salem* differ from that on the *Chester*, which introduces another inevitable complication in any attempt at comparison, by weight, of coal consumed.

While the water-measuring apparatus was so arranged that on the steam-consumption tests underway, the steam used by the main propelling machinery could be separately determined from the steam used by the auxiliaries in operation as a whole, it was considered desirable, nevertheless, to fix by actual test and under varying conditions the steam necessary for each individual auxiliary. These tests, which constitute Group I, were made on each vessel while at the Boston Navy-Yard, and the results recorded are more particularly described under "Tests of auxiliary machinery."

Group II covers evaporative boiler tests on the *Chester* and *Salem* with the vessels in port. It may be remarked that similar tests on the boiler plant of the *Birmingham* were considered unneccessary since this plant is identical in all important particulars with the *Salem's* installation. These consisted of a series of four tests on each vessel, using the two after boilers, which latter connect to and are the only boilers delivering gases of combustion to the after smokestack. On the *Salem* these consisted of forced draft tests with closed fireroom under air pressures equivalent, respectively, to one-half, $1\frac{1}{4}$, $2\frac{1}{2}$, and 4 inches of water. A series of tests was made on the *Salem* at the various air pressures stated, but the board considered these unsatisfactory. A second series was run at a later date, therefore, under more favorable conditions and with satisfactory results. Only the results of the second series of tests, however, are tabulated in this report. The *Chester's* tests covered one at natural draft, with others at one-half, $1\frac{1}{4}$, and $2\frac{1}{2}$ inches of water, respectively, the aim being to simulate conditions of steaming in service and to approximate to the same coal consumption per square foot of grate in the boilers of each vessel.

Standardization runs (Group III) were conducted under the supervision of the Board of Inspection and Survey, and results are plotted on plates 85 (*Birmingham*), 86 and 87 (*Salem*), and 88 (*Chester*).

Group IV comprises tests on all ships, measuring exhaust steam of condensation from the main and auxiliary machinery, with vessels at sea, and at various speeds, from about 10 knots up to the maximum. In these tests the exhaust steam entering the main condensers was, after condensation, measured in the main measuring tanks; furthermore, if found desirable, the amount entering each condenser could be separately determined, thus making it possible to compare the steam consumptions of the starboard and port main turbines or engines. Exhaust steam from all auxiliaries in operation,

including drains, discharge from traps, etc., was delivered to the auxiliary condenser and, after condensation, discharged to the auxiliary measuring tank. The total boiler output, therefore, in steam during any test, amounted to the sum of the weights of water collected in the main and auxiliary tanks. No attempt to measure or weigh the coal used on these trials was made. On the *Birmingham*, 10 steam-consumption tests, as above described, were run in all, and the data collated is given in Tables 33 to 42. Two series of tests were made on the *Salem;* the first set, which included 10 tests, furnished clue to damage of starboard main turbine blading, which eventually resulted in a complete overhaul at the contractors' works (Fore River Shipbuilding Company), and repair of the injured parts. The discovered defects are mentioned more at length later in this report. As, however, the repairs referred to were not undertaken until after completion of the (V) competitive coal-consumption trials, and the twenty-four hours' (VI) full-power run, the data obtained on these trials are given in full, in addition to the results on 14 other steam-consumption tests forming the second series, which latter were made when the machinery of the *Salem* had been placed in good condition. Steam-consumption trials on the *Chester*, embracing tests of the 4, 5, and 6 turbine combinations, at various speeds, and numbering 29 in all, were carried out. Of these, the results of the first 5 are at variance with results of later similar tests in the series. This may have been due to the fact that the 5 tests referred to were the first steam-consumption tests undertaken of the main and auxiliary machinery, as a whole, and it is probable that inaccuracies resulted from inexperience in operating the machinery in connection with measuring apparatus, the difficulties of which were greatly accentuated in the beginning by unsatisfactory working of the main lift pumps, which later were put in proper condition. The steam-consumption results of these 5 tests are regarded as unreliable, but have been tabulated in Tables 66 to 70 for other data.

Group V comprises data collated from sea runs of the three vessels in company, extending over periods of ninety-six, fifty, and ninety-eight hours, respectively, and at speeds of about 10, 15, and 20 knots. On these trials the same grade of coal, loaded at the naval coaling station, Bradford, R. I., was used on all ships, and the amount burned by each vessel was carefully determined. As it was desired that the mean displacement for each trial and for each ship should be as nearly as possible 4,000 tons, calculation was made prior to every coaling, of the amount necessary to be taken aboard, so that, making due allowance for the coal required to propel the ship to the open sea before each trial began, and, as well, that necessary for the trial itself, the displacement, when one-half of the trial was finished, would approximate to the figure stated. It is evident, then, that this involved starting each trial at a greater displacement than 4,000 tons, and ending with a displacement below that figure. All trials were begun off the harbor of Newport, R. I., and finished close to that port. Each ship maintained, at all times, very closely the revolutions indicated as necessary by the standardization curve for speeds of 10, 15, and 20 knots, respectively, and at no time were the vessels widely separated, being always in sight of one another, and, of course, subject to the same weather conditions. Particular attention was given to secure accurately the number of cubic feet of coal

consumed, but, in addition, such other data were taken as would enable the steam consumption of all machinery in use to be approximately and separately figured from results of steam-consumption tests previously made. Prior to these trials, water-measuring apparatus had been removed from all ships.

The full power trials (VI), extending over a period of twenty-four hours, were undertaken with the view of demonstrating the highest sustained sea speed, for the time mentioned, of these vessels. As in all other sea trials, the aim was to have each vessel at an average displacement for the trial of 4,000 tons, as nearly as could be figured. The coal for all ships was of the same grade and procured at the Bradford coaling station. As previously stated, the *Salem* and *Chester* finished this trial, but the *Birmingham* was obliged to discontinue, after twelve hours, on account of looseness of I. P. crosshead (starboard engine), which caused violent vibrations and liability to further damage in event of continuing the trial.

A brief description of the hull and machinery of these vessels is given below.

PRINCIPAL HULL DATA.

The hulls (pl. 1) of all vessels, except in minor particulars, are alike. Steel is used throughout, and the outside plating generally is on the raised and sunken strake system. The frames, spaced 36 inches apart, are generally of channel section, 6 by $2\frac{13}{16}$ by $2\frac{13}{16}$ by 13.3 pounds.

The outside plating, below the load water line, consists of 15-pound plating, reduced to 12-pound at the ends, and of 12-pound plating elsewhere. Flat keel plates, about 36 inches in width, are in two thicknesses; the inner of $17\frac{1}{2}$ and the outer of 20 pound plate. The vertical keel, about 39 inches in depth, is of 15-pound plate. Garboard and sheer strakes are $17\frac{1}{2}$ pounds, reduced to 15 pounds at the ends. Nickel-steel protection (80-pound plate) of variable width, extends fore and aft in wake of engine, boiler, and dynamo rooms.

The weight of hull, including nickel-steel protection, but without machinery, coal, stores, outfit, armament, and ammunition, etc., is approximately 2,015 tons. Principal dimensions are:

Length between perpendiculars, feet	420
Length over all, feet and inches	423—2
Length on L. W. L., feet	420
Breadth, molded, feet and inches	46—8
Breadth, extreme, feet and inches	47—$0\frac{3}{4}$
Ratio of length to beam	8. 97
Draft (official contract trial, 3,750 tons displacement), feet and inches	16—9
4,000 tons displacement, feet and inches	17—$4\frac{1}{2}$
4,710 tons displacement (about fully loaded), feet and inches	19—$2\frac{3}{4}$
Displacement per inch at mean draft (16 feet 9 inches), tons	31. 07
Area of midship section (3,750 tons displacement), square feet	566
Area of L. W. L. section (3,750 tons displacement), square feet	12, 960
Wetted surface section (3,750 tons displacement), square feet	19, 900
Coefficient (at 3,750 tons displacement):	
Block	. 40
Midship	. 72
L. W. L. plane	. 66
Coal bunker capacity in tons (43 cubic feet per ton):	
Birmingham	1, 395. 3
Salem	1, 388. 3
Chester	1, 407

MACHINERY INSTALLATIONS.

Main propelling machinery, of 16,000 I. H. P., at maximum power, of the twin-screw, reciprocating type, for the three scout cruisers, was designed by the department. Proposals, however, for the construction of these vessels were invited under two classes, viz, first, for hull and machinery in accordance with department's plans and specifications; second, in general accordance with the department's plans and specifications, but on bidder's design of machinery, preference to be given, other things being equal, to a turbine installation.

After examination of the bids submitted, contracts were awarded on the department's design of hull, as follows: To the Fore River Ship and Engine Building Company, Quincy, Mass., two vessels—the *Birmingham* and *Salem*—at a contract price of $1,556,000 each; the first named to have the department's design for propelling machinery, and the latter to be equipped with Curtis marine turbines. Contract for the remaining vessel, the *Chester*, was awarded to the Bath Iron Works, Bath, Me., at $1,688,000; the propelling installation to be of the Parsons marine type.

Both contractors were required to guarantee, under penalty, and in addition to various other trials, a successful sea trial of four hours' duration at an average speed of not less than 24 knots on a displacement of not less than 3,750 tons.

A summary of the important data of the propelling machinery of the three vessels follows:

BIRMINGHAM.

[Plate 2.]

The engines, which turn the propellers outboard, are of the vertical inverted 4-cylinder, direct-acting, triple-expansion type, with unjacketed cylinders, placed in two water-tight compartments, and operating twin screws. Each engine was designed for an indicated horsepower of 8,000 at 200 revolutions per minute, with a steam-chest pressure of 250 pounds per gage. Beginning forward, the order of the cylinders is, forward L. P., H. P., I. P., and after L. P. The forward L. P. and H. P. cranks are opposite, as are the I. P., and after L. P., the second pair being at right angles with the first. All main valves are of the piston type, worked by double-bar Stevenson links. There is one piston valve for the H. P. and two each for the I. P. and L. P. cylinders.

Engine framing is of forged steel, cylindrical columns, trussed by forged-steel stays. All crank, thrust, line, and propeller shafting is hollow, and shafts, piston, and connecting rods, and working parts generally, are of forged steel:

CYLINDERS.

Number for each engine	4
H. P. diameter, inches	28¼
I. P. diameter, inches	45
F. L. P. diameter, inches	62
A. L. P. diameter, inches	62
Diameter of piston rods, inches	6
Stroke of all pistons, inches	36

PROPELLERS (MANGANESE BRONZE).

Number of blades	3
Diameter, feet and inches	12—6
Pitch, (as set) mean, feet and inches	15—3
Pitch, adjustable, from feet and inches	14—6 to 16—6
Ratio of diameter to pitch	1. 22
Area, projected, square feet	40. 8
Area, helicoidal, square feet	49. 4
Area, disk, square feet	122. 7

CONDENSERS.

There is one main condenser in each engine room cylindrical in form, the important tube data being as follows:

Number of tubes	4, 085
Diameter (outside) inches	$\frac{5}{8}$
Thickness, B. W. G	#16
Length between tube sheets, feet	12
Spacing (between centers) inches	$\frac{15}{16}$
Cooling surface, square feet	8, 000

In the forward engine room, there is an auxiliary condenser of 600 square feet cooling surface. There is also an auxiliary condenser for the dynamo plant, located in the dynamo room, having a cooling surface of 200 square feet.

FEED-WATER HEATERS.

There is a feed-water heater in each engine room, located on the discharge side of the main feed pumps, with 600 square feet of heating surface, composed of 679 $\frac{5}{8}$-inch tubes.

BOILERS.

Plates 8 and 10 show the boiler arrangement on the *Birmingham* and *Salem*, which are similar, and give the most important data of the boiler installations on these vessels. It should be particularly noted that the products of combustion leave at the back of the furnaces, and on their way to the uptakes, make a double passage across the tubes. In the *Chester's* boilers, shown on plates 9 and 10, only a single pass is made by the gases of combustion, which enter the tubes near the front of the boiler furnaces.

FORCED-DRAFT BLOWERS.

The closed fireroom system is used, there being 6 independent Sturtevant fans installed, 2 to each boiler compartment, 66 inches in diameter, 24-inch width at tips of blades, of the double inlet type, with 42-inch intakes. Each fan is directly driven by a double-acting, 2-cylinder, simple, horizontal engine (6 by 6) with cranks at 180°, the steam admission and emission for both cylinders being controlled by 1 piston valve.

AUXILIARIES.

Table A gives principal dimensions of the auxiliaries on all 3 vessels.

Table A.

Identification letter.	Auxiliaries.		Birmingham.					Salem.					Chester.				
	Name.	Type.	No.	Diameter of cylinders.				No.	Diameter of cylinders.				No.	Diameter of cylinders.			
				No.	Steam.	Water.	Stroke.		No.	Steam.	Water.	Stroke.		No.	Steam.	Water.	Stroke.
a	Main air pumps	Double acting	2	2	12″	22″ air.	18″										
		Single acting	2										2	2	14″	35″ air.	21″
b	Augmenter												2	⅜-inch nozzle.			
c	Wet vacuum pump	Two stage vertical centrifugal.						2	Two stage 27″ rotor turbine pump, rotor 12″ d.								
d	Rotary dry vacuum	Vertical with fly wheel						2	1	10″	24″ air.	18″					
e	Main circulating	Centrifugal	2	2	8″ 16″	36″ runner.	8″	2	2	8″ 16″	36″ runner.	8″	2	2	10″ 16″	42″ runner.	10″
f	Main feed pumps	Vertical simplex	3	1	15″	10″	15″	3	1	15″	10″	15″	3	1	15″	10″	15″
g	Reserve feed pumps	do	1	1	6″ Blake.	4″ duplex.	7″	1	1	3½″	4″	4″	1	1	10″	9″	12″
h	Oil circulating pumps	Horizontal duplex	2	1	2″	2¼″ oil.	2¾″										
		Vertical simplex											2	1	10″	9″ oil.	12″
		do						4	1	4″	5″ oil.	6″					
i	Fire and bilge pumps	do	2	1	10″	8½″	12″	2	1	10″	8½″	12″	2	1	10″	9″	12″
k	Auxiliary air and circulating pumps.	Simplex horizontal combined.	1	1	6″	10″ 10″ air.	12″	1	1	6″	10″ 10″ air.	12″	1	1	6″	10″ 10″ air.	12″
l	Auxiliary feed pumps	Vertical simplex	3	1	9″	6″	12″	3	1	9″	6″	12″	3	1	9″	6″	12″
m	F. and B. on ash ejector	do	3	1	12″	8″	12″	3	1	12″	8″	12″	3	1	12″	8″	12″
n	Forced draft blowers	Horizontal double	6	2	6″	66″ fan.	6″										
		Double upright											6	2	5″	84″ fan.	5″
		Horizontal double						6	2	6″	66″ fan.	6″					
					Horizontal.					Horizontal.					Vertical.		
o	Distiller circulating pumps	Simplex	2	1	10″	10″	12″	2	1	10″	10″	12″	2	1	10″	9″	12″
p	Evaporator feed pumps	Vertical simplex	1	1	4½″	5″	6″	1	1	4½″	5″	6″	1	1	4½″	5″	6″
r	Distiller fresh water pumps	do	1	1	4½″	5″	6″	1	1	4½″	5″	6″	1	1	4½″	5″	6″

TABLE A—Continued.

Identification letter.	Auxiliaries.		Birmingham.					Salem.					Chester.				
	Name.	Type.	No.	Diameter of cylinders.				No.	Diameter of cylinders.				No.	Diameter of cylinders.			
				No.	Steam.	Water.	Stroke.		No.	Steam.	Water.	Stroke.		No.	Steam.	Water.	Stroke.
s	Air compressors	Westinghouse	2	2	11″	11″	12″	2	2	11″	11″	12″	2	2	11″	11″	12″
t	Dynamo plant	Cross compound	3	2	7½″ 12″	.....	8″	3	2	7½″ 12″	.....	8″	3	2	7½″ 12″	.....	8″
u	Dynamo A. and C. pump	Simplex horizontal combined.	1	1	6″	8″ 8″ air.	12″	1	1	6″	8″ 8″ air.	12″	1	1	6″	8″ 8″ air.	12″
v	Ice machinery	1-ton Allen dense air	2	1	8″	Compressor, 6½″. Expander, 5½″.	8″	2	1	8″	Compressor, 6½″. Expander, 5″.	8″	2	1	8″	Compressor, 6½″. Expander, 5½″.	8″
w	Steering engine	Simple vertical	1	2	12″	.....	10″	1	2	12″	.....	10″	1	2	12″	.....	10″
x	Anchor engine	do	1	2	12″	.....	10″	1	2	12″	.....	10″	1	2	10″–12″	Compound.	10″

SALEM.

[Plate 3.]

The *Salem* is driven by outboard turning twin screws, the propelling machinery consisting of 2 Curtis marine reversible impulse turbines, 1 on each shaft, designed to develop 8,000 B. H. P., at 350 revolutions per minute with 250 pounds (gage) steam-chest pressure. The machinery is arranged in 2 water-tight compartments, separated by an athwartship bulkhead, the starboard screw being operated by the forward turbine.

Each turbine has a pitch diameter of rotating wheels of about 120 inches (center to center of buckets), and for the ahead motion, consists of 7 stages; each wheel is fitted with 3 rows of buckets, except in the first stage, which contains 4 rows. For backing, there are 2 stages, the wheels or rotors being mounted on the shaft in the same casing as the go-ahead stages.

Each turbine consists of a cast-iron cylindrical shell (for strength, cast-steel is used in construction of the casing of the two first stages) divided by dished cast-iron or cast-steel diaphragms into separate compartments or stages. The moving buckets are mounted on the periphery of separate wheels in each stage, and all wheels are secured to and carried by a hollow steel shaft extending the length of the turbines. Where this shaft passes through the diaphragms, bronze bushings are provided, with small clearances, in order to prevent appreciable steam leakage from stage to stage, and where the shaft extends through the heads carbon-packed gland boxes are fitted to prevent steam leaking out at the ahead end, or air leaking in at the back end. These boxes are supplied with live steam between carbon packing which makes an effective seal against air leaking in at the back end, and which, unless checked, would cause the vacuum to be lowered. Gland-box packing, on the ahead end, is required to withstand the pressure existing in the first stage only, in order to prevent steam leakage into the engine rooms; and, as this pressure is never excessive, under any condition of operation, no serious practical difficulty arises in guarding against such leakage. A drain, or "leak off," from these gland boxes leads to the fourth stage. The ahead steam chest contains 20 expanding nozzle openings to the first stage wheel, 17 of which are controlled by valves, the 3 remaining openings being without valves and consequently always open. When in operation, sufficient nozzle valves are opened to give the desired speed, the throttle valve being left wide open in order to secure full pressure in the steam chest. Variation in speed within limits may be made by throttle-valve regulation, but usually this is accomplished by opening or closing of nozzle valves.

Steam chests are of separate steel castings attached to each turbine-casing head. The astern chest contains the same number of nozzles as the ahead chest, but only 8 are provided with valves. In maneuvering, nozzle valves are left open and speed controlled by throttle valves.

First stage nozzles are of the expanding type, for the steam in its passage to the first revolving wheel undergoes too great a drop in pressure to permit the use of the parallel-flow type, which latter is used for all other stages, the drop in pressure being much less. Diaphragm plates have openings cast in them to allow passage of steam through to the nozzles.

Drain pipes are fitted between each stage and the succeeding one, in order that condensed steam in any stage may pass to the next one of lower pressure. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. Generally, however, this ejector is used only in getting underway, being closed off when the turbines have assumed normal working conditions.

A longitudinal section of the *Salem's* turbine, with important dimensions, is shown on plate 4. An outline of the machinery arrangement with main steam piping is shown on plate 3.

PROPELLERS.

Prior to official government acceptance trials of the *Salem* the contractors fitted and tried four different sets of propellers. The design which gave the best results (cast solid) and which was finally used has the following principal dimensions:

Number of blades	3
Diameter, feet and inches	9 – 6
Pitch, feet and inches	8 – 8½
Ratio, diameter to pitch	1.09
Area, projected, square feet	36.8
Area, helicoidal, square feet	43.7
Area, disk, square feet	70.9

CONDENSERS.

Two exhaust pipes connect each main turbine to a cylindrical, horizontal surface condenser, located in each engine room. In connection with each condenser, there is installed one Blake, vertical, dry-air pump, having 1 steam cylinder and 1 air cylinder, each double acting, operating on the same crank with fly wheel between the two cranks. There is also a centrifugal wet vacuum pump for each condenser, of the 2-stage type, directly connected and driven by a small 2-stage (27 inch) Curtis turbine.

Number of tubes (each condenser)	4,448
Diameter of tubes (outside) inches	⅝
Thickness, B. W. G	#16
Length, feet and inches	13 – 2
Cooling surface, square feet (each condenser)	9,460
Diameter exhaust, inlets (2), inches	42

There is an auxiliary condenser in the forward engine room and a condenser for the dynamo plant, both of which are similar in size to those heretofore described for the *Birmingham*.

BOILERS, FEED-WATER HEATERS, AND BLOWERS.

The size and location of boilers, feed-water heaters, and blowers are the same as on the *Birmingham*.

AUXILIARIES.

A list of auxiliaries will be found in Table (A).

CHESTER.

[Plate 6.]

The propelling machinery of the *Chester*, designed for a working steam pressure at the turbines of 250 pounds (gage) and about 507 revolutions per minute for the contract speed of 24 knots, consists of

Parsons marine steam turbines, driving 4 independent shafts, each shaft being fitted with 1 propeller. The installation comprises 6 ahead turbines in all, 2 of which—the H. P. and I. P. cruising—are for use at low powers in order to secure economy. There are also 2 backing turbines, located in each of the L. P. casings, and it is to be observed that this arrangement permits reversal only of the 2 inboard shafts in maneuvering.

Referring to plate 6, it will be seen that the outboard shafts (Nos. 1 and 4) are operated entirely by main H. P. turbines; the starboard inboard (No. 2) by the L. P. (starboard) turbine and H. P. cruising turbine, and the port inboard (No. 3) by the L. P. (port) turbine and the I. P. cruising.

The turbines may be used in any of three ways in propelling the vessel: First, for speeds up to about 18 knots, all 6 turbines are put in operation, since this combination results in a smaller quantity of steam being required than with either of the other two combinations. Steam is then admitted initially into the H. P. cruising turbine, exhausts into the I. P. cruising turbine, and from the latter splits and is conveyed through separate pipes to each of the main H. P. turbines. From these latter, steam is exhausted into the L. P. turbines, and finally into the condensers. Second, for speeds beyond the capacity of the 6-turbine combination, and up to about 23 knots, the 5-turbine combination is used. Steam is admitted initially to the I. P. cruising turbine, passing thence to the two main H. P. turbines, and from each of them to the connected L. P. turbine. The H. P. cruising turbine revolves idly in a vacuum. Third, for highest speed, only the 4 main turbines are used, steam being admitted initially to each main H. P. turbine, exhausted into the L. P's and then into the condensers. Both cruising turbines revolve idly in a vacuum, or may be disconnected by means of shaft couplings. It is to be understood, of course, that the 4-turbine combination may be used for any speed up to the full power, just as the 5-turbine combination may be employed for any speed up to its limit.

In any of the three arrangements, increase or reduction of power is effected by throttling, though in the 6-turbine combination, a by-pass is fitted between the first and second expansion which may be used, within limits, for this purpose. All maneuvering is done with the 4 main turbines only, and in event of either the 6 or 5 turbine combinations being in use when a signal is received to go astern, a quick shift is necessary to cut out cruising turbines. This simply involves closing and opening proper valves, and need not require a greater interval of time than is usual in reversing machinery of the reciprocating type.

For the removal of water, all turbines have a connection to a system of drain piping leading to the main condensers. The outlet from each turbine is controlled by a valve which is always opened when warming up, and also on the cruising turbines when underway in case these are not in use for propulsion.

Shaft gland boxes (12 in all) are bolted to forward and after casing heads of each turbine. In order to prevent leakage, a steam pressure of about 1 pound above the atmosphere is usually maintained in gland boxes when cruising, through a system of piping installed for the purpose. Plate 7 shows construction of gland boxes as installed on H. P. and L. P. turbines.

As shown on plate 6, the machinery is installed in two compartments, separated by an athwartship water-tight bulkhead, the two cruising turbines, in addition to the starboard main H. P. and L. P. being located in the forward engine room. For the ahead motion the inboard shafts turn outboard, and the outboard shafts turn inboard.

The turbine cylinders are parted horizontally and the halves bolted together. The lower half is cast with extensions, box shaped forward and aft, for retaining the journal and thrust bearings.

Important data of turbine dimensions, stages, blading, etc., is shown below:

	Diameter of rotor drum.	Length of rotor drum.	Diameter of cylinder for each stage.						
			First.	Second.	Third.	Fourth.	Fifth.	Sixth.	Seventh.
	Inches.	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*
H. P. cruising	60	36	60¾	61	61¼	....	....	....	....
I. P. cruising	49	60.5	51¼	51¾	52.5	....	....	....	....
Main H. P. (port and starboard)	42	103.5	43¾	44.5	45.5	47	49	52	....
L. P. (port and starboard)	65	53⅜	70	72	75	79	79	79	79
Astern (port and starboard)	50	43	51	52	54	54	54	....	....

Stages.	Expansions.	Rows.	Heights.	Pitch.	Clearances.
H. P. CRUISING.			*Inches.*	*Inches.*	*Inches.*
First	First	12	⅜	⅞	0.03
Second	Second	12	½	⅞	.035
Third	Third	12	⅝	1	.04
I. P. CRUISING.					
First	First	15	1⅛	1¼	.04
Second	Second	15	1 7/16	1¼	.04
Third	Third	15	1¾	1 5/16	.045
MAIN HIGH PRESSURE.					
First	First	12	⅞	1¼	.03
Second	Second	12	1¼	1¼	.035
Third	Third	12	1¾	1 5/16	.04
Fourth	Fourth	12	2½	1⅜	.045
Fifth	Fifth	12	3½	1 7/16	.05
Sixth	Sixth	10	5	1 13/16	.05
MAIN LOW PRESSURE.					
First	First	5	2½	1 5/16	.055
Second	Second	5	3½	1⅜	.06
Third	Third	4	5	1 13/16	.07
Fourth	Fourth	3	7	1¾	.08
Do	Fifth	3	7	1 15/16	.08
Do	Sixth	4	7	2 5/16	.08
Do	Seventh	4	7	2 1/16	.08
ASTERN TURBINES.					
First	First	6	½	1⅛	.045
Second	Second	6	1	1⅜	.05
Third	Third	6	2	1⅜	.06
Do	Fourth	6	2	1½	.06
Do	Fifth	6	2	1½	.06

PROPELLERS. (MANGANESE BRONZE, CAST SOLID.)

Number of propellers	4
Number of blades (each propeller)	3
Diameter, feet	6
Pitch, feet	6
Ratio of diameter to pitch	1
Area, projected, square feet	17.02
Area, helicoidal, square feet	19
Area, disk, square feet	28.27

CONDENSERS.

The main condensers, one in each engine room, are located abreast the L. P. turbines. They are cylindrical, of the surface-condenser type, and each is supplemented by a vacuum augmenter installed beneath the condenser. The augmenter consists of a steam siphon, drawing air from the main condenser and discharging it to the air-pump suction. The siphon discharge passes through a small surface condenser (which latter is supplied with circulating water by connection to the salt-water side of the main condenser) in order to condense steam of the siphon jet. The main air pump has a direct suction from the condenser through a pipe having a water seal, holding a head of water equal to the difference in pressure produced by the augmenter jet. It is figured that an increase in vacuum of approximately 1 inch of mercury results from the use of the augmenter.

The principal tube data of each main condenser are:

Number of tubes		5,630
Diameter (outside), inches		5/8
Thickness, B. W. G		#18
Length of tubes, feet and inches		10–0 1/4
Spacing (between centers), inches	upper half	1 x 1 1/8
	lower half	1 x 1
Cooling surface, square feet		8,999
Exhaust inlets (rectangular), feet and inches		3–9 x 4–6

There is one auxiliary and one dynamo condenser, located in the forward engine room and dynamo room, respectively, both similar and of the same dimensions as those previously described for the *Birmingham* and *Salem*.

FEED-WATER HEATERS.

A cylindrical feed-water heater, composed of 630 5/8-inch tubes and containing about 600 square feet of heating surface, is located near the forward bulkhead of each engine room on the discharge side of the main feed pumps. The heating agency is steam from the auxiliary exhaust line, which enters the shell at the top, circulates around the tubes over a system of baffles, and drains through a trap to the main drain line. The tubes, through which the feed water passes on its way to the boilers, contain twisted brass strips, held in place by perforated plates over the tube sheets to retard the flow and thus insure efficient heating. A small air coil is fitted in the lower water chest to prevent the heater from becoming air bound. This coil is connected to the inside of the heater shell, near the bottom, and drains to the main condenser.

BOILERS.

The principal dimensions and arrangement of boilers are shown on plates 9 and 10. As previously pointed out, the gases of combustion make but a single pass across the tubes in their exit to the uptakes. It is to be noted, furthermore, that, although the grate surface is the same in the boiler installations of all three vessels, the *Chester's* boilers contain but 32,040 square feet of heating surface, as against 37,992 square feet in the boiler plants of the *Birmingham* and *Salem.*

FORCED-DRAFT BLOWERS.

The system of forced draft used on the *Chester* is similar to that on the other scout vessels. It is of the closed fire-room type, there being 2 blowers (6 in all) 84 inches in diameter for each boiler compartment. Each blower is driven by a vertical two-cylinder (5 by 5) simple double-acting steam engine, with one piston valve to each cylinder. The blowers are arranged in pairs near the fore and aft center line of ship, with a connecting shaft between each two fans, so that both fans run at the same speed, and in event of accident to one engine the remaining engine may be utilized to operate both fans.

AUXILIARIES.

A list of auxiliaries is given in Table A.

MACHINERY WEIGHTS.

Weight of propelling machinery, shafting, bearings, and propellers is as follows:

	Tons.
Birmingham (2 shaft reciprocating engine arrangement)	234.49
Salem (2 shaft Curtis turbine arrangement)	254.80
Chester (4 shaft Parsons turbine arrangement)	207.38

Weight per horsepower (the maximum power of main engines or turbines for two consecutive hours during either steam or coal consumption trials) is:

	Pounds.
Birmingham (16134), per I. H. P.	32.55
Salem (18070), per S. H. P.	31.58
Chester (20004), per S. H. P.	23.22

The weight of appendages (auxiliaries) to the above propelling machinery, including main condensers, air pumps, circulating pumps, and dry vacuum pumps or vacuum augmenters is as follows:

	Tons.
Birmingham	45.13
Salem (fitted with dry-vacuum pumps)	67.63
Chester (fitted with vacuum augmenters)	58.19

The boilers, fittings, smoke pipes and uptakes weigh as follows:

	Dry.	Wet.
	Tons.	*Tons.*
Birmingham (12 Fore River boilers, 4 smoke pipes)	262.51	294.14
Salem (12 Fore River boilers, 4 smoke pipes)	262.61	294.24
Chester (12 Nomand boilers, 4 smoke pipes)	272.70	307.01

Weight of boilers with fittings and water in pounds per square foot of heating surface is:

	Pounds.
Birmingham	13.10
Salem	13.09
Chester	17.64

The total weight of machinery installation, including propelling machinery and appendages, auxiliary machinery, piping, boilers, and fittings, smoke pipes and uptakes, lagging and clothing, flooring, ladders and gratings, fittings and gear, stores, tools, and spare parts carried on board, and pipes, etc., connecting to machinery not under the cognizance of the Bureau of Steam Engineering, is as follows:

	Dry.	Water.	Total.
	Tons.	*Tons.*	*Tons.*
Birmingham	760.82	53.15	843.97
Salem	853.85	55.11	908.96
Chester	735.87	64.82	800.69

The weight per horsepower (main propelling machinery only) of all machinery (wet) is:

	Pounds.
Birmingham (16134), per I. H. P	117.18
Salem (18070), per S. H. P	112.68
Chester (20004), per S. H. P	89.67

MEASURING APPARATUS.

A type plan of water-measuring apparatus installed, together with necessary lift pumps and piping, is shown in outline on plate 11. For the propelling machinery the apparatus is composed essentially of two cylindrical tanks, one for each engine room, divided by a central diaphragm. These tanks were located on the spar deck and emptied by gravity into two distributing tanks situated immediately below on the engine-room grating. Water from each main condenser was pumped into its own feed tank in the usual way and from there elevated to the measuring tanks by means of lift pumps. It is to be noted that the central diaphragm of the measuring tanks served to divide each into two separate, similar compartments, thus, in reality, providing two measuring tanks of about 6,100 pounds capacity for each engine room. These tanks were of conical construction at the top and bottom, and each compartment terminated at the top in a cylindrical nozzle 11½ inches in diameter.

After measurement the water was dropped to the distributing tanks, which were connected by an equalizing pipe, and had a combined capacity of about 24,700 pounds. Suction pipes led from distributing tanks to feed pumps, thus providing means of returning water of condensation to the boilers. Some of the first tests on the *Chester* were made without any connection between distributing tanks, and this was found unsatisfactory, the tanks not being large enough to take, at all times, the full capacity of a liberated measuring tank. In addition to this, the lift pumps in the beginning gave considerable trouble, necessitating the use of one main lift pump only,

and thus requiring the two main feed tanks to be connected by opening the equalizing valve. Subsequently the two distributing tanks were connected to overcome this difficulty. The water of condensation from the auxiliaries was, after measurement, delivered into the forward distributing tank, and any make-up water required during the tests was pumped from the double bottoms directly into this tank. It should be noted that the auxiliary measuring appliances, embracing tanks, piping, and lift pumps, constitute a system entirely distinct and separate from the main measuring apparatus.

The discharge from each lift pump terminated in a swinging goose neck which permitted readily shifting from one compartment to the other of each tank. The shell of the main measuring tanks was carried above the head, thus forming an open-topped receptacle in which any overflow from the measuring tanks was caught and returned to the feed tanks by means of drain pipes.

Steam used by the various auxiliaries, heaters, steam traps, drains, etc., was condensed by the auxiliary condenser and the water of condensation delivered into an auxiliary tank installed as part of the test outfit. This tank, of rectangular section, was divided into two compartments, with an approximate capacity of 1,130 pounds of water each, and by means of an auxiliary lift pump could be readily emptied into the forward distributing tank as desired.

The dynamo plant, composed of three 32-kilowatt units, was provided with its own condenser and pumps, and the air pump delivery was so arranged that the water of condensation could be delivered either directly into the auxiliary measuring tank or to the E. R. auxiliary condenser. As the auxiliary exhaust line of piping was also connected to the dynamo plant, it was possible to utilize the dynamo condenser to assist in condensing the steam from the auxiliaries in operation in case the auxiliary condenser became too small in condensing surface to properly handle the auxiliary exhaust steam at high powers.

CALIBRATION OF MEASURING TANKS.

All measuring tanks, prior to installation on shipboard, were carefully calibrated on platform scales at the Boston Navy-Yard. Each tank was filled with water at 100° temperature and the weight of contained water ascertained for each inch rise in height. Glass water gauges, graduated brass scales, and thermometers were attached to each tank for use during steam consumption trials.

Three photographs, marked, respectively, "plate 13," "plate 14," and "plate 15," show these tanks during process of calibration.

I. TESTS OF AUXILIARY MACHINERY.

After installation of auxiliary measuring tanks, and while the vessels were at the Boston Navy-Yard, a series of tests was made to establish the steam consumption of the various individual auxiliaries on each ship. The method adopted was to isolate and condense the exhaust steam from the auxiliary under test, using the forward engine room auxiliary condenser, and to ascertain the weight of the resulting water of condensation. To facilitate securing the data sought, temporary atmospheric exhausts were rigged to such auxiliaries as were

likely to be run either constantly or intermittently during the tests, and which ordinarily exhausted into the auxiliary exhaust line. These included an auxiliary boiler feed pump, steam cylinder operating air and circulating pumps of auxiliary condenser, ice machines, etc. Dynamo installations, as previously pointed out, were provided with independent condensing plants, so that no special provision to handle exhaust steam from these was necessary.

In the three ships the auxiliary exhaust line, in a fore-and-aft direction, runs almost parallel to the center line, with suitable branch connections to each of the various auxiliaries. This line of piping was used to convey exhaust steam from the auxiliary being tested to the engine room auxiliary condenser, from which, in the form of water, it was delivered by the air pump to the auxiliary measuring tanks. Plate 12, figure 3, shows the construction of these auxiliary tanks, and it will be seen then, for each ship, but one tank, of rectangular section, provided with a central water-tight diaphragm, was required. The division of each into two separate compartments constituted, in reality, two measuring tanks of about equal capacity, into either of which the water of condensation could be directed, as desired, by a swinging gooseneck.

No special provision, other than fitting certain appliances (gages, counters, and indicator gears), was made to prepare any auxiliary for test, a superficial examination being alone considered necessary when the auxiliary appeared to be in satisfactory working order. It was the aim to operate each auxiliary under conditions met with in service, but examination or special fitting of steam cylinder packing to influence steam economy was not attempted. Counters were generally fitted in all cases where not permanently installed, to determine accurately the revolutions, or double strokes, during the test; in addition, steam gages were installed on steam chests, and in some tests, such, for example, as those of feed, flushing, and ash-ejector pumps, gages were also fitted to indicate discharge pressures. Moreover, to assimilate actual service conditions, a back pressure of about 6 pounds above the atmosphere was maintained in the auxiliary exhaust line, during all tests, by manipulating a regulating valve governing inlet to auxiliary condenser, the pressure noted being that usually carried when exhaust steam from the auxiliaries is utilized in the feed-water heaters, as is the custom in cruising.

It was usual, before the commencement of an auxiliary test, to subject the system as a whole to a preliminary test for tightness. This merely involved inspection of the air-pump discharge, after starting auxiliary condenser, the absence of water issuing therefrom being accepted as indicating not only that the auxiliary exhaust line was clear, but as well that the condenser was free from salt water leaks. The auxiliary to be tested was next started, but before beginning to record data, sufficient time—usually one-half hour or more—was allowed to elapse to insure normal working conditions. During this period the water of condensation was permitted to run directly through one of the two compartments of the measuring tank to the feed tank below. At the instant of beginning the test the swinging goose neck, attached to the air-pump discharge, was shifted and directed into the other empty compartment of the measuring tank, the gate valve at the bottom of this having been previously closed.

The water of condensation was thus collected, and by aid of a graduated scale and water-gage glass, attached to each compartment, the height in inches, as the tests progressed, as well as full tanks, could be noted and recorded. Opportunity offered, as each compartment was being filled, to prepare the remaining one for use, so that tests could be carried on for any length of time, and without interruption, by alternately shifting goose-neck connection of air-pump discharge from one to the other as the compartments filled. In some tests, as will be observed from the tables, it was not considered necessary to collect an entire tank of condensed water, while others were continued to include several tanks. Boiler and steam-chest pressures and pressure in auxiliary exhaust line, with the data necessary to compute power, are recorded in appropriate columns in the various tables (2 to 32) of auxiliary tests.

It will be apparent from a survey of the tabulated data that essentially the same mode of procedure was adopted in all auxiliary tests. At the start of each the water of condensation was directed into one compartment of the measuring tank, and as the test advanced, the elapsed time to fill the various heights (usually by progressive intervals of 5 or 10 inches) up to "tank full," together with the temperature of water at each reading, was recorded. From tank calibration tables, making due allowance for temperature—which correction generally was immaterial—the weight of collected water up to any height was readily acsertained. Regularity in time interval to fill equal increments in height (each tank compartment being closely of the same rectangular section up to about 45 inches) usually indicated regularity in steam consumption of the auxiliary under test. This, however, was not necessarily the case, as it was noticed, particularly when the measuring tank filled slowly and therefore the quantity of steam being condensed was comparatively small, that gulps of water were frequently delivered at irregular intervals. The cause of this may be traced to pockets in the exhaust line of piping or possibly to failure of the air pump to deliver small quantities of water with unvaried regularity.

Tables 2 to 32 present the important data recorded in all auxiliary tests, and from these, performance curves, more particularly described below, have been laid down. In constructing these curves it has been assumed that dry saturated steam was furnished at all times, which assumption, insofar as concerns the quality of steam generated, seems to be sustained by results of evaporative boiler tests, which are recorded later in this report. It may be pointed out further, in this connection, that care was taken to always have ample boiler power in use in order to guard against forced evaporation, and that the auxiliary steam line, from which all auxiliaries received their supply, was, in addition to being well lagged, provided with traps, etc., for proper drainage.

To assist in analyzing results of auxiliary tests, graphical illustrations on cross-section paper have been plotted as follows:

First. A number of curves (plates 16 to 32) using steam consumption per hour as abscissæ, with ordinates representing revolutions (or double strokes) per minute, multiplied by the weight of a cubic foot of steam at the average steam-chest pressure.

Second. Plates 33 to 42 with abscissæ representing steam consumption per hour and ordinates I. H. P. developed during the tests.

Third. Plates 43 to 51 with abscissæ representing water per hour per I. H. P. in pounds, and ordinates I. H. P.

The first set is believed to be more useful than the other two, and has proved of substantial and unquestionable value in differentiating the steam used by individual auxiliaries on main steam-consumption trials, where the condensed exhaust steam from all auxiliaries, as a whole, was measured; also, in the coal-consumption trials, to calculate approximately the steam used by the various auxiliaries in operation. It is to be observed that plotted points of the first group seem to lie generally in a straight line, and the conclusion has been reached that this invariably follows for those auxiliaries with constant cut-off and which remain in a constant working condition. The data for such points as do not fall fairly upon the line, it is believed, are in slight error, due, no doubt, to the difficulty of accurately reading gages attached to steam chests, which, on many tests, fluctuated widely. In such cases it was customary to record as the true reading the mean of the arc of vibration, determined by carefully reducing or throttling the inflow of steam to the gages through plug cocks at the bottom.

Auxiliary machinery, which is entirely independent of the main motive power and which is but slightly, if at all, dependent in steam consumption upon speed of propulsion, may be grouped under two headings, as follows: First, those in which actual measurable mechanical work is done, such, for example, as dynamos, ice machines, various pumps, etc. For many of these, horsepower water-consumption curves have been plotted, while for others, like steering engines, because of intermittent use, any such curve would necessarily be of questionable value. Under the second heading may be classed those appliances which, although requiring steam for their operation, do not, strictly speaking, convert the energy of steam into mechanical work, and which, moreover, as regards steam consumption, are dependent, to a large extent, upon local conditions. Evidently any attempted estimate of power for such appliances, based upon steam used, would be valueless.

In comparing trial performances of the three vessels, as will appear later, it has been necessary to convert indicated into shaft horsepower for the main propelling engines of the *Birmingham*, but a similar conversion, as applied to the auxiliaries, has not been attempted in any case. While for the total developed power of each ship this involves adding shaft horsepower of the propelling machinery and indicated power of the auxiliaries, which is undesirable from the standpoint of unlike units, nevertheless this method seems more satisfactory than entering into an intricate calculation as to the relation between the two units for each auxiliary, which, however carefully done, could only be regarded finally as an approximation.

Table A gives principal dimensions of the various auxiliaries, and it will be noticed that in the machinery equipment of each vessel there are several of the same size and make. Tests of economy were undertaken only on one of each size, and the results obtained are considered equally applicable to all of the same size when similarly used.

For convenience of reference, identification letters have been placed opposite the several columns of the table, and these, in conjunction with a designating letter for each ship, are to be found on data sheets

and plates of the various auxiliary tests. For example, the letters a–B are to be understood as designating the main air pump of the *Birmingham:* a–C, the main air pump of the *Chester;* e–S, the main feed pump of the *Salem;* the letters f–B–S–C would be used to signify auxiliary feed pumps of all vessels.

The auxiliaries installed on each vessel, with respect to location, may be classed under the following three heads: 1, engine-room auxiliaries; 2, fireroom auxiliaries; and 3, other or outside auxiliaries. In commenting further upon results of auxiliary machinery tests, this classification will be observed.

1. ENGINE-ROOM AUXILIARIES.

MAIN AIR PUMPS.[a]

(a–B–C.)

Table 2 shows the data collected, and plates 16 and 33 plotted performance curves, of four tests of the after main air pump of the *Birmingham.* This pump is of the Blake featherweight variety, double-acting (both steam and water), simplex type, with two steam and two water cylinders. Each steam piston gives motion to its own water piston, and the arrangement is such that one half of the pump may be in operation while the remaining half is at rest. When cruising at low powers, and to effect steam economy, it is customary to use but one side of the pump. In tests 1, 2, and 3 one steam cylinder only was in use; in test 4, both steam cylinders were in operation at the same speed. During all tests the line of piping, which collects the discharge from all traps, was open to main condenser; the air pump handled no other water; main injection and outboard delivery valves were wide open, but circulating pump was not in operation. Pump made full (18 inches) stroke on all tests. In estimating and apportioning to each auxiliary the weight of steam used on the various sea trials a consumption of 186 pounds per indicated horsepower per hour has been taken for this pump.

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[a] In air pumps of the type installed on the *Birmingham*, where there are no valves to open behind the pump plunger, a good vacuum of probably 28.5 (0.75 pound absolute) or better, must always exist behind the suction side of the plunger until it passes the edge of the slot where the vacuum will drop to that of the condenser. The pressure on the ahead or discharge side of the plunger will start with the absolute pressure in the condenser. As the piston rises, the pressure on the discharge side will increase, according to Mariotte's law. Owing to the water present, the rise in temperature will be very small and most of the vapor will condense as the volume decreases, since its pressure can not exceed that due to its temperature. The air present, however, will be compressed and its pressure will rise until the discharge valves open. With the temperature of the condensed steam in the condenser remaining constant, the vacuum can change only with an increase or decrease in the amount of air present, which causes an increase or decrease of load on the plunger.

In case of either the *Birmingham's* or *Chester's* pumps, the actual load on the plunger will be the absolute pressure on the discharge side at any point of the stroke, minus the constant pressure behind the plunger, that is, on the suction side. The part of the load on the discharge side is the same for the two types of pumps. But the part of the load on the suction side, which is to be subtracted from the load on the discharge side to get the net load, is different in the two pumps.

In the *Birmingham's* pump the suction side pressure would be a perfect vacuum in case of no clearance at the beginning of the stroke, but under working conditions the pressure is that resulting from temperature of water on top of the discharge valves, evaporated as the plunger recedes, and which is constant throughout the stroke, no allowance being made for the slight variation in pressure due to a drop of temperature on account of the evaporation of the water. As stated above, 0.75 pound absolute

Tests of the *Chester's* main air pump are tabulated in Table 3, and performance curves plotted on plates 17 and 33. These pumps are of the Blake, beam, vertical type, with two steam (double-acting) cylinders, having a common steam chest and two (single-acting) water cylinders. Main trap discharge line was open to main condenser during all tests, as well as main injection and discharge valves; circulating pump was not in use. Stroke of pump during tests was 18¼ inches. The same steam consumption as stated for the *Birmingham's* air pump, per unit of power per hour, has been taken for this pump in calculations relating to individual steam expenditures of auxiliaries on sea trials.

AUGMENTORS.

(b–C)

For the purpose of further adding to the vacuum produced by the air pumps, an augmentor is installed on the *Chester* in connection with each main condenser. When the ship is under way these are in constant use, the steam thus expended being practically the same in amount for all speeds. Tests in port, with the propelling machinery stopped and extending over a period of two and one-half hours, were carried out to determine the weight of steam used, the water of condensation being delivered to and measured in the main feed tanks. With an average steam pressure of 184 pounds (gage) and steam-supply valve open to the augmentors the usual number of turns, the weight per hour was found to be: Forward, 904; and, aft, 992 pounds. The theoretical weight of steam, by Napier's formula, which would flow through a nozzle of the same diameter as in the augmentors, as well as that actually found by test, is graphically represented on plate 18. On (IV) steam-consumption trials it will be apparent that all steam used by the augmentors necessarily forms part of the measured water of condensation from the

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has been assumed as the maximum of this constant pressure. In the *Chester's* pump, the part of the load on the suction side (which is constant) is the absolute pressure in the condenser minus the pressure lost in passing the air, vapor, and water through the foot valves. As 28.5 inches vacuum (0.75 pound absolute pressure) is assumed as the maximum pressure on the suction side of the *Birmingham's* pump, the load on the *Chester's* pump will be the same, provided the condenser has a vacuum of 28.5 (0.75 pound absolute) plus a pressure necessary to force the air, vapor, and water through the foot valves. For every increase of absolute pressure in the condenser over the assumed pressure of 0.75 pound, plus the resistance in foot valves, there will be a corresponding reduction of load on the suction side of the *Chester's* pump, as compared with the *Birmingham's* pump, by the increase in pressure.

In conducting auxiliary tests of air pumps, they were run at various speeds and with vacuums varying from 21 to 23 inches on the *Birmingham*, and 14 to 23 inches on the *Chester*. It was not possible, however, to make any of the tests with the main propelling machinery in operation, and hence under usual service conditions as to vacuum; so, while the steam consumption, as determined for chest pressures and revolutions, is regarded as quite accurate and applicable to service conditions, the water per I. H. P. can not be so regarded. For this reason no curves for water consumption and horsepower have been drawn. The water per I. H. P. has been assumed to be 186 pounds for all the tests of both the *Chester* and *Birmingham*. While this would actually vary somewhat, the total I. H. P. of the pumps is so small that the error will affect the total result but slightly, and in no way affect the actual steam consumption of the pumps at the various speeds, since that has been determined from the revolutions and chest pressure, as already stated.

main propelling machinery. An equivalent indicated horsepower, based upon a steam consumption of 50 pounds per horsepower per hour, has been assigned to each augmentor in estimating power of auxiliaries on sea trials.

WET (CENTRIFUGAL) AND DRY (ROTARY) VACUUM PUMPS.

(c–d–S)

Because of the importance, with turbine machinery, of maintaining a high vacuum at all times, a wet and dry vacuum pump form part of each main condenser equipment of the *Salem*. The first mentioned is of the centrifugal vertical type, driven by a Curtis 2-stage turbine, which is provided with three steam nozzles, each of tapering rectangular (0.0375 square inch area at the smallest part) cross section. Two of the three steam nozzle openings are fitted with control valves, the remaining one being constantly open. The pump speed is regulated by a governor, and steam, after passing the throttle, is admitted to the turbine through one, two, or three nozzles, as found requisite to handle efficiently the water of condensation resulting from entry of exhaust steam into the condenser. No power test on this type of pump was attempted; tests to fix steam economy with one and two nozzle openings—discharge from various traps only entering the main condensers—are recorded in Table 4, from which data (and also from results of other tests made by the Fore River Shipbuilding Company) the graphical representation on plate 19 has been constructed. It may be remarked here that on all steam-consumption tests of the *Salem's* propelling machinery, the steam necessary to operate these pumps could not well be separated from that used by the main turbines; for, although proper connections to the auxiliary exhaust line were installed, unsatisfactory working resulted with vessel under way, when so operated against a back pressure above the atmosphere. The exhaust from each pump, therefore, was directed into its main condenser. In calculations relating to power of auxiliaries on sea trials, an equivalent indicated horsepower has been assigned to each wet vacuum pump, based upon a steam consumption of 50 pounds per hour per I. H. P.

For the removal of air, etc., each main condenser is provided with a rotary dry vacuum pump, consisting of one air cylinder, operated by a single-cylinder steam engine. Results of tests of this auxiliary are tabulated in Table 5, and performance curves drawn on plates 20, 33, and 43.

MAIN CIRCULATING PUMP ENGINES.

(e–B–S–C)

A compound engine operates a centrifugal pump to supply cooling water to each main condenser. These installations on the *Birmingham* and *Salem* are alike in all particulars, except that the engine shaft of the latter carries a bilge pump worked by a geared crank. H. P. cylinder diameter and stroke of the *Chester's* engine are somewhat larger than similar dimensions of the engine fitted in the other two ships, permitting in consequence the development of greater power. Tables 6, 7, and 8 give, respectively, the results of tests of

these auxiliaries, from which plates 21, 34, and 44 have been laid down. In all tests the main injection and outboard delivery valves were wide open.

OIL-CIRCULATING PUMPS.

(h-S-C)

To insure proper lubrication of main turbine bearings and thrust blocks, oil-circulating pumps, operating on a closed system, are installed. The *Salem* is provided with two such pumps in each engine room, but under ordinary cruising conditions the operation of one pump in connection with each turbine is sufficient. Oil is continuously supplied from closed tanks installed in the upper engine-room hatches, to the various bearings, under a gravity head of about 20 feet, and drained therefrom to coolers located near the pumps. The purpose of the coolers is to reduce the temperature of oil flowing to them, each being arranged by suitable connections to the main circulating pumps for circulation of cooling water. Each pump has a suction to the coolers with a discharge to the supply tank.

The *Chester* has but one oil-circulating pump in each engine room, either or both of which may be used as desired on the oiling system. Oil is supplied to the bearings under pressure (about 10 pounds), and the arrangement of coolers, etc., in essential particulars is similar to that described on the *Salem*.

No steam-consumption tests were carried out on these pumps. A test, however, was begun on the *Chester*, and discontinued after it became apparent that the amount of steam used under ordinary conditions of working was extremely small, and could only be accurately determined by continuing the test over an abnormal time interval. Steam used by these pumps on main steam and coal consumption trials, as well as power, has been estimated from results of other pump tests; a figure of 200 pounds per I. H. P. per hour has been assumed in all cases.

MAIN FEED PUMPS.

(f-B-S-C)

In the original design, two main feed pumps for each ship, of the size shown in Table A, were estimated to be of sufficient capacity to supply the boiler installations with the requisite quantity of feed water when developing maximum power. Upon official trials, however, a maximum speed and power greater than that planned, especially with the turbine-propelled ships, was developed, making it desirable, at least viewed from the standpoint of safety, to provide an additional feed pump. This led eventually to fitting a third feed pump on each vessel of the same size as the other two, although these pumps were not finally installed until just prior to the coal-consumption runs. In the meantime, and before the main steam-consumption tests were undertaken, each vessel was provided with a temporary additional feed pump for emergency use.

Tables 9, 10, and 11 contain results of the various tests carried out, the pumps in all cases making full strokes. Performance curves are plotted on plates 22, 35, and 45. It should be observed that

during all tests, in order to assimulate service conditions, the pumps delivered water against a pressure approximating that generally carried in the main feed line when steaming at sea. This was accomplished by arranging the pumps to draw from double bottom compartments and redelivering the fresh water to the same place, a valve on the delivery side of pumps being regulated to maintain the desired pressure.

FIRE AND BILGE PUMPS.

(i–B–S–C)

These tests were conducted with fire and bilge pumps drawing water from and returning it to the sea, under varying discharge pressures, as shown on Tables 12, 13, and 14. Performance curves are plotted on plates 23, 36, and 46. In all tests pumps made full stroke.

ENGINE ROOM AUXILIARY CONDENSER.

(k–B–S–C)

One cylindrical, horizontal, auxiliary condenser is located in the forward engine room of each vessel. Attached pumps (air and circulating) are horizontal and operated by a single steam cylinder. Tables 15 and 16 contain results of tests of this auxiliary, a performance curve being plotted on plate 24.

2. FIRE-ROOM AUXILIARIES.

AUXILIARY FEED PUMPS.

(l–B–S–C)

In tests of these pumps fresh water was pumped from and returned to the double bottom compartments, the discharge pressure being regulated in the same manner and for the same reason as heretofore explained under "Main feed pumps." Results are shown on Tables 17 and 18, and curves of performance on plates 25, 37, and 47. Pumps made full stroke in all tests.

FIRE AND BILGE PUMPS (USED IN EJECTING ASHES).

(m–B–S–C)

In each fireroom there is installed a fire and bilge pump, which may be used, in conjunction with a hopper arrangement, for ejecting ashes. At such times, to insure successful working, it is essential to carry a comparatively high pump discharge pressure. Tests recorded in Tables 19 and 20 were made under normal conditions of service with the pumps subjected to this use. Performance curves are shown on plates 26, 38, and 48. Pumps were so regulated as to make designed stroke on all tests.

FORCED-DRAFT BLOWER ENGINES.

(n–B–S–C)

On the *Birmingham* and *Salem* the forced-draft blower installations consist of six separate directly-connected blowers, two to each fireroom compartment, driven by independent, 2-cylinder, simple engines. Table 21 contains data of five tests (one blower and engine) on the *Birmingham,* operating under different steam-chest pressures, free escape of air supplied the fireroom being permitted through discharge opening of the blower at rest. In these tests care was taken to maintain at all times precisely the steam-chest pressure tabulated.

Table 22 contains the results of the six tests on the *Salem*, with two blowers in use, under air pressures varying from one-half to 4 inches of water, the fireroom being closed air-tight, and escape of air governed, by opening boiler furnace doors. It was the aim to carry a constant air pressure in each of these tests, the blower engines being speeded or slowed, as required, to accomplish this end.

Eight tests on the *Chester's* forced-draft blower engine, recorded in Table 23, were carried out in the same way as noted for the *Salem*. Performance curves are shown on plates 27, 39, 40, and 49.

3. OUTSIDE AUXILIARIES.

DISTILLER CIRCULATING PUMPS.

(o–B–S–C)

Two distiller circulating pumps, either or both of which may be used, in addition, for flushing purposes, are installed in each ship. Tests recorded in Tables 24 and 25 were with the pumps delivering sea water to the flushing main only, the evaporating plant being shut down. With the latter in operation it is the ordinary practice to use but one pump to supply cooling water to the distillers, while the other operates on the flushing line. Performance curves are plotted on plates 28, 41, and 50.

EVAPORATING PLANT.

(o–p–r–B–S–C)

The fresh-water distilling plant on each vessel consists of four evaporators (single effect) and four distillers, having a combined capacity of about 16,000 gallons potable water per day. Four pumps are fitted in connection with each plant, as follows: Two distiller circulating (o), one evaporator feed (p), and one fresh water (r).

Tests of four hours' duration were carried out on each ship, with two evaporators and accessories in use, the object being to ascertain the weight of fresh water distilled per pound of boiler steam used, with the plant operating under normal service conditions. The condensed steam collected included that discharged from evaporator coils, as well as the exhaust from various (one distiller circulating, evaporator feed and fresh water) pumps. Results are recorded in Table 26, and a graphical representation shown on plate 29.

ELECTRIC PLANTS.

(t–B–S–C)

Each vessel is provided with three 32-kilowatt General Electric generating sets, compound wound, of the multipolar type, and each driven by a direct-connected compound engine. In addition to lighting the vessel, current is supplied for electrical operation of a number of auxiliaries, such, for example, as deck winches, hull ventilation fans, workshop machinery tools, etc. An auxiliary condenser, similar in type to that fitted in the forward engine room, but smaller in dimensions, is installed for the exclusive use of each dynamo plant.

Tables 27, 28, and 29 contain data covering tests of one engine on each vessel under various electric loads. Indicator cards were secured on almost all tests, but the results from those taken on the *Birmingham* and *Salem* exhibited such obvious inconsistencies (due probably to inaccurate indicator springs or faulty indicator connections) as to justify their rejection. Cards from the *Chester's* engine apparently give fairly consistent and reliable results, and the power as determined from these has been used for all ships in plotting curves on plates 30, 42, and 51.

Tests to establish the quantity of steam necessary for operation of the air and circulating pumps of dynamo auxiliary condensers (u–B–S–C) were not made, as the steam cylinder driving these pumps is of the same diameter and stroke as fitted to auxiliary condensers in engine rooms.

REFRIGERATING PLANTS.

(v–B–S–C.)

Two vertical, 1-ton Allen dense-air (each having 1 steam, 1 compressor, and 1 expander cylinder) ice machines are installed on each ship, and tests of these are contained in Tables 30 and 31, with a graphical representation of steam consumption per hour shown on plate 31. As proper gear was lacking for the motion, no indicator cards were secured from the steam cylinder.

STEERING ENGINES.

(w–B–S–C.)

A Williamson steam steering engine (2-cylinder, simple, horizontal) is installed on the *Birmingham* and *Salem*. The *Chester* is provided with a Hyde steering engine of similar type and principal dimensions, the power, in all ships, being transmitted to the rudder head by a right-and-left-hand screw shaft.

Table 32 (pl. 32) gives the weight of steam wasted per hour by the steering engine of each ship (due to leakage) with throttles open (change valves in midposition) but no movement of the engines. Results are also tabulated (main propelling machinery not in use) when helm is slowly shifted, at precisely two-minute intervals, 3° to starboard, then the same angle to port, and immediately afterwards brought to rest amidships.

A summary of results of all auxiliary tests is shown in Table B.

TABLE B.—*Synopsis of results of auxiliary machinery tests.*

1.	2.	3.	4.	5.	6.	7.	8.	9.	
Reference letter and ship.	Auxiliary.	No. of table and test.	Duration of test.	Steam-chest pressure gage.	Revolutions or double strokes per minute.	I. H. P. (average).	Condensed exhaust steam. (Pounds.) Per hour.	Condensed exhaust steam. (Pounds.) Per I. H. P. per hour.	Remarks.
			H. m. s.						
a-B		2-1	1 30 21	50.0	26.90	4.0	753.1	188.2	1 cylinder only in use.
a-B		2-2	1 31 22	44.0	21.10	3.5	744.8	212.8	Do.
a-B		2-3	2 09 40	42.0	14.30	2.8	524.7	187.4	Do.
a-B		2-4	0 40 46	43.0	26.90	7.3	1,663.1	227.8	Both cylinders in use.
a-C	Main air pumps	3-1	0 35 39	105.0	28.90	21.0	3,824.1	182.1	Vacuum main condenser, 23.1.
a-C		3-2	0 28 52	82.0	21.90	16.9	2,361.0	139.7	Vacuum main condenser, 21.0.
a-C		3-3	0 42 49	73.0	15.90	14.1	1,593.2	113.0	Vacuum main condenser, 18.1.
a-C		3-4	1 15 43	58.0	9.60	7.8	900.9	115.5	Vacuum main condenser, 14.0.
c-S	Turbine wet vacuum pumps	4-1	2 13 40	199.0	1,314.00	....	508.1	....	1 nozzle open to steam turbine.
c-S		4-2	1 35 21	150.0	1,292.00	....	712.3	....	2 nozzles open to steam turbine.
d-S		5-1	0 49 30	85.0	60.10	20.7	1,373.3	66.3	
d-S	Dry vacuum pumps	5-2	0 42 05	71.0	80.10	26.1	1,615.4	61.89	
d-S		5-3	0 59 48	70.0	45.20	14.2	1,136.8	80.0	
e-B		6-1	0 41 25	104.4	239.00	48.9	1,637.0	33.5	
e-B		6-2	1 18 33	59.3	177.00	21.5	866.2	40.3	
e-B		6-3	2 23 15	32.2	110.00	6.0	416.7	69.4	
e-B		6-4	2 36 00	15.7	58.00	1.8	219.3	121.8	
e-S		7-1	1 52 14	46.0	87.50	3.6	302.6	84.1	
e-S	Main circulating pumps	7-2	1 24 33	82.0	155.10	15.9	795.5	50.0	Main injection and discharge valves wide open during all tests.
e-S		7-3	0 25 20	173.0	275.80	81.0	2,643.0	32.6	
e-S		7-4	0 16 37	193.0	406.20	161.6	3,990.0	24.7	
e-C		8-1	0 39 12	128.0	233.20	142.9	3,482.5	24.35	
e-C		8-2	0 35 28	100.0	186.90	87.0	1,922.0	22.0	
e-C		8-3	1 00 20	73.0	145.00	37.0	1,135.0	30.6	
e-C		8-4	1 57 55	35.0	76.80	9.3	411.1	44.2	

TABLE B.—*Synopsis of results of auxiliary machinery tests*—Continued.

1.	2.	3.	4.	5.	6.	7.	8.	9.	
Reference letter and ship.	Auxiliary.	No. of table and test.	Duration of test.	Steam-chest pressure gage.	Revolutions or double strokes per minute.	I. H. P. (average).	Condensed exhaust steam. (Pounds.) Per hour.	Condensed exhaust steam. (Pounds.) Per I. H. P. per hour.	Remarks.
			H. m. s.						
f-B		9–1	0 14 32	198.0	50.00	66.4	4,661.0	70.2	Discharge pressure gage, 200 pounds.
f-B		9–2	0 21 10	183.0	30.00	46.4	3,214.5	69.3	Discharge pressure gage, 212 pounds.
f-S		10–1	0 37 47	165.0	12.80	19.8	1,775.3	89.7	Discharge pressure gage, 300 pounds.
f-S		10–2	0 22 57	203.0	24.80	41.7	2,917.0	70.0	Do.
f-S		10–3	0 40 00	211.0	34.30	52.8	4,185.6	79.4	Do.
f-S	Main feed pumps	10–4	0 24 30	240.0	45.30	72.1	5.446.7	74.55	Do.
f-S		10–5	0 17 12	223.0	34.30	52.4	3,934.9	75.1	Do.
f-C		11–1	0 28 37	153.0	13.00	18.1	2,381.7	131.6	Discharge pressure gage, 268 pounds.
f-C		11–2	0 21 02	187.0	24.50	41.7	3,229.4	77.4	Discharge pressure gage, 265 pounds.
f-C		11–3	0 28 30	254.0	33.00	59.6	4,766.4	80.0	Discharge pressure gage, 286 pounds.
i-B		12–1	1 08 20	75.0	55.00	9.5	999.2	105.2	Discharge pressure gage, 30 pounds.
i-B		12–2	0 35 55	99.0	53.20	11.1	1,147.7	103.4	Discharge pressure gage, 45 pounds.
i-B		12–3	1 10 56	46.0	54.00	5.6	581.1	103.8	Discharge pressure gage, 15 pounds.
i-S		13–1	1 41 10	49.0	41.90	2.6	398.5	153.3	Discharge pressure gage, 10 pounds.
i-S	Engine-room fire and bilge pumps	13–2	1 49 50	56.0	39.70	3.8	612.4	161.2	Discharge pressure gage, 20 pounds.
i-S		13–3	1 34 05	87.0	40.30	7.0	714.9	102.1	Discharge pressure gage, 40 pounds.
i-C		14–1	0 39 12	120.0	54.90	13.3	1,740.2	130.8	Do.
i-C		14–2	1 24 32	85.0	56.10	8.3	807.1	97.2	Discharge pressure gage, 20 pounds.
i-C		14–3	1 05 12	56.0	40.30	4.1	420.6	102.5	Do.
i-C		14–4	0 45 36	79.0	55.90	6.6	755.2	114.4	Discharge pressure gage, 15 pounds.
k-B	Engine-room auxiliary air and circulating pump.	15–1	1 35 11	67.0	60.00	....	504.9	....	Power not indicated.
k-S		15–1	1 39 00	....	44.80	1.6	266.1	166.3	
k-C		16–1	1 22 35	100.0	59.90	4.7	500.5	106.5	

l-B	Fireroom auxiliary feed pumps	17-1	1 45 44	156.0	13.30	4.8	643.5	134.1	Discharge pressure gage, 311 pounds.
l-S		17-1	0 35 10	258.0	35.00	20.3	1,907.5	94.0	Discharge pressure gage, 300 pounds.
l-S		17-2	0 47 40	225.0	25.00	14.3	1,409.8	98.6	Do.
l-S		17-3	0 15 15	190.0	15.00	7.8	893.8	114.6	Do.
l-C		18-1	0 52 39	187.0	29.70	14.7	1,295.7	88.14	Do.
m-B	Fire and bilge on ash ejector	19-1	0 32 26	180.0	36.00	25.6	2,101.6	82.1	Disharge pressure gage, 280 pounds.
m-S		19-1	0 15 58	263.0	50.00	40.2	4,193.6	104.3	Discharge pressure gage, 265 pounds.
m-S		19-2	0 24 25	202.0	36.60	25.0	2,742.3	109.7	Do.
m-S		19-3	0 40 55	166.0	22.20	13.9	1,639.4	117.9	Do.
m-C		20-1	0 25 30	176.0	36.30	31.4	2,674.0	85.1	Discharge pressure gage, 262 pounds.
n-B	Forced-draft blowers	21-1	1 18 30	125.0	380.00	.....	1,738.7	.....	No air pressure in fireroom.
n-B		21-2	0 43 30	110.0	360.00	.....	1,569.7	.....	Do.
n-B		21-3	1 06 15	95.0	310.00	.....	1,030.6	.....	Do.
n-B		21-4	0 45 00	75.0	260.00	.....	758.7	.....	Do.
n-B		21-5	0 36 50	124.0	379.00	.....	1,853.8	.....	Do.
n-S		22-1	0 12 06	149.5	486.00	.....	5,509.0	.....	4 inches of air pressure.
n-S		22-2	0 17 34	106.0	399.50	32.2	3,811.6	59.2	2½ inches of air pressure.
n-S		22-3	0 53 31	54.0	223.50	6.7	1,255.7	93.7	1 inch of air pressure.
n-S		22-4	0 53 44	41.5	165.00	3.2	753.7	117.8	½ inch of air pressure.
n-S		22-5	0 13 43	125.0	427.00	35.9	4,872.9	67.9	3 inches of air pressure.
n-S		22-6	0 28 00	76.5	305.00	16.6	2,391.5	72.0	1½ inches of air pressure.
n-C		23-1	0 26 10	106.0	393.00	21.7	2,604.8	60.0	4 inches of air pressure.
n-C		23-2	0 41 46	70.0	313.00	11.7	1,631.9	69.7	2½ inches of air pressure.
n-C		23-3	1 05 03	42.0	190.00	3.3	636.4	96.4	1 inch of air pressure.
n-C		23-4	1 08 03	52.1	206.00	4.7	1,001.6	106.4	Do.
n-C		23-5	0 32 09	87.0	327.00	15.9	2,120.2	66.7	2½ inches of air pressure.
n-C		23-6	0 19 15	143.0	424.00	34.4	3,534.5	51.4	3.9 inches of air pressure.
n-C		23-7	0 56 28	65.0	223.00	6.7	1,208.2	90.2	1 inch of air pressure.
n-C		23-8	0 21 15	130.0	382.00	27.6	3,204.6	58.1	2½ inches of air pressure.

TABLE B.—*Synopsis of results of auxiliary machinery tests*—Continued.

1.	2.	3.	4.	5.	6.	7.	8.	9.	
Reference letter and ship.	Auxiliary.	No. of table and test.	Duration of test.	Steam-chest pressure gage.	Revolutions or double strokes per minute.	I. H. P. (average).	Condensed exhaust steam. (Pounds.) Per hour.	Condensed exhaust steam. (Pounds.) Per I. H. P. per hour.	Remarks.
			H. m. s.						
o-B	Distiller circulating pumps (used for flushing).	24–1	2 55 00	47.0	30.70	3.5	779.0	222.57	Discharge pressure gage, 25 pounds.
o-S		25–1	1 01 47	73.0	53.30	9.5	1,088.6	114.6	Discharge pressure gage, 30 pounds.
o-S		25–2	1 05 38	63.0	35.30	5.9	827.3	140.2	Do.
o-S		25–3	1 11 30	45.0	20.40	3.1	368.4	118.8	Results of test unreliable.
o-S		25–4	1 10 07	88.0	33.00	7.6	959.3	126.2	Discharge pressure gage, 50 pounds.
o-S		25–5	1 39 52	45.0	34.50	3.8	673.5	177.2	Discharge pressure gage, 20 pounds.
o-C		24–1	1 51 50	62.0	54.60	6.1	610.0	100.0	Discharge pressure gage, 12 pounds.
o. p. r.–B	Evaporators and distilling plant	26–1	4 00 00				a3,190.8	b2,608.3	Two evaporators in use.
o. p. r.–S		26–1	4 00 00				a4,140.0	b2,980.3	Do.
o. p. r.–C		26–1	4 00 00				a4,999.2	b3,687.8	Do.
t-B	Dynamos	27–1	1 08 45	108.0	400.00	20.4	989.6	48.5	I. H. P. figured from similar tests on the Chester.
t-B		27–2	1 02 40	107.0	400.00	29.9	1,085.7	36.3	
t-B		27–3	0 52 04	108.0	400.00	39.2	1,306.7	33.3	
t-B		27–4	0 43 40	108.0	400.00	46.6	1,558.1	33.4	
t-B		27–5	0 34 00	109.0	395.00	55.7	1,994.1	35.8	
t-B		27–6	0 42 41	108.0	400.00	20.8	1,594.0	38.3	
t-B		27–7	1 23 03	108.0	400.00	20.6	819.3	39.8	
t-S		28–1	1 46 50	110.0	400.00	20.4	629.0	30.8	Do.
t-S		28–2	1 26 00	110.0	400.00	29.9	781.4	26.1	
t-S		28–3	1 02 08	111.0	400.00	39.2	1,081.5	27.6	
t-S		28–4	0 51 18	112.0	400.00	46.6	1,310.0	28.1	
t-S		28–5	0 46 28	112.0	400.00	55.7	1,446.2	26.0	
t-C		29–1	0 53 05	100.0	382.00	48.5	1,285.5	26.5	

t-C		29-2	0 44 43	100.0	381.00	53.8	1,525.6	28.4	
t-C		29-3	0 56 52	100.0	386.00	40.5	1,207.7	29.8	
t-C		29-4	2 20 30	96.0	388.00	20.4	485.6	23.8	Steam consumption unreliable.
t-C		29-5	1 50 35	100.0	386.00	30.1	616.9	20.5	Do.
t-C		29-6	1 17 55	101.0	387.00	....	875.5	29.1	
t-C		29-7	1 43 27	103.0	393.00	....	659.4	32.3	
v-B	Ice machines	30-1	0 50 55	73.5	118.00	....	670.5	....	Power not indicated.
v-S	Ice machines	30-1	1 09 08	49.0	95.85	....	486.1	....	Do.
v-C	Ice machines	31-1	2 26 46	62.0	96.20	....	464.4	....	Do.
v-C	Ice machines	31-2	1 01 27	67.0	97.70	....	555.0	....	Do.
w-B	Steering engines	32-1	0 40 10	....	....	....	1,026.0	....	Engine at rest.
w-B	Steering engines	32-4	0 45 07	....	....	....	1,511.0	....	
w-S	Steering engines	32-3	1 22 00	....	....	....	492.0	....	Do.
w-S	Steering engines	32-6	0 43 53	....	....	....	919.0	....	
w-C	Steering engines	32-2	1 16 20	....	....	....	542.0	....	Do.
w-C	Steering engines	32-5	0 52 10	....	....	....	793.0	....	

[a] Total steam used per hour by entire plant.

[b] Pounds of water distilled per hour.

II. EVAPORATIVE BOILER TESTS.

The *Birmingham* and *Salem* were built by the same contractors, the Fore River Shipbuilding Company, and, as previously pointed out, the boiler installations are alike. Work of constructing these boilers was sublet to the Sterling Boiler Company, of Barberton, Ohio, upon designs prepared by the shipbuilding company. The boilers of the *Chester*, which are of the same general type as fitted in the other two vessels, differ in amount and disposition of tube-heating surface, path of gases of combustion, and other particulars, as will be seen by reference to plate 10. They were built at the works of the Bath Iron Works, the contractors for the vessel.

On each vessel the boiler plant consists of 12 water-tube boilers of the express type, installed in three separate water-tight compartments, and in all vessels the general arrangement, number and location of firerooms, number and height of smoke pipes, etc., is practically identical.

A series of evaporative tests were made on the *Salem* and *Chester*, using the two after boilers (designated by the letters L and M), with a view of comparing boiler efficiencies under different rates of combustion. The tests were carried out (in port) under as nearly as possible actual service conditions, and the boilers fired by the ships' force. It was considered unnecessary to repeat tests of this character on the *Birmingham*, because of similarity of boiler installations on this vessel and the *Salem*.

Before beginning each series of tests, and after the heating surfaces had been properly cleaned and examined, a test water pressure (boilers cold) was applied in order to make sure that no leaks existed. Further cleaning of the boilers was done after each test, generally at night, as thoroughly as that was possible without hauling fires or allowing them to die out.

The main water-measuring apparatus—installed primarily to secure data on steam-consumption tests of the propelling machinery—furnished an easy and accurate method of determining the weight of water pumped to the boilers. Feed water was supplied to the forward measuring tank (fig. 1, pl. 12) and afterwards dropped to the lower or distributing tank (fig. 2, pl. 12), from which latter, through a suction connection to the feed pump, it was delivered to the boilers. All tanks prior to installation in the ships were carefully calibrated, and the weight pumped to the boilers during any test was readily calculated from its volume and temperature.

The ships' main delivery feed line was made use of to supply water to the boilers, and this was carefully examined for leaks at intervals as the tests progressed. Only the use of a comparatively short length of this line, however, was necessary, as by closing certain valves in the after fireroom the delivery of feed water forward of the boilers under test was not possible.

During tests on the *Salem*, feed water was pumped directly to the boilers from the lower or distributing tank, its temperature being somewhat raised prior to measurement, by drains blowing into the main feed tank. On the *Chester*, the feed pump discharged through the forward heater, and the water on its passage to the boilers, was raised in temperature by exhaust steam from the auxiliaries in use.

Steam generated by the boilers under test was supplied to the

main steam line, and so much utilized as was necessary to operate port auxiliaries; any excess was disposed of either through specially rigged piping directed overboard (*Salem*) or delivered to the main condenser (*Chester*) through bleeder pipe connections.

Moisture in the steam was determined by Barrus throttling calorimeters, one of these instruments being fitted to each boiler and connected to main stop valve chambers on boiler side of valves. The distance of calorimeters from boiler drums was about 12 inches.

Coal was weighed on platform scales in the firerooms, and at the beginning and end of each test, fires were carefully leveled so as to have, as nearly as could be judged, the same amount on the grates. It may be stated here that in planning these tests it was the intention to use the same kind and grade of coal in all, but on account of the movement of the vessels this proved inconvenient. The two series were made, therefore, with Georges Creek (*Salem*) and Eureka (*Chester*) coal, the vessels being temporarily at the Boston and New York navy-yards, respectively, when the tests were carried out.

Considerable ash and cinder were blown out of the smoke-pipes in all tests except No. 4 (natural draft) on the *Chester*. Because of this, determinations of weight of ashes hauled from the ash pans were considered valueless, and these weights, in consequence, have been figured from results of chemical analyses of the coal used.

Temperature of smoke-pipe gases were taken (about 35 feet above grates) by mercurial and electric pyrometers. In tabulating results, however, readings only of the mercurial pyrometer are recorded, except in test No. 3 on the *Chester*. In this test, flame was observable, at frequent intervals, through the aperture into which the pyrometers fitted, a circumstance which did not occur during other tests, and the electric pyrometer only was used on account of high temperature.

Analyses of flue gases on each test were made by Orsatt apparatus, at hourly periods on the *Salem* and at half-hourly intervals on the *Chester*. The average of all such analyses for each test is recorded in the tables.

A summation of the data of all tests is given in Tables C and D, and it will be noted, from a comparison of results, that the boilers installed on the *Salem*, under approximately similar conditions as to consumption of coal, show a higher thermal efficiency than the type on the *Chester*. This, unquestionably, is attributable, in part, to the larger percentage of tube-heating surface in the *Salem's* boilers, but more particularly to the tortuous passage traversed by the gases of combustion.

Graphical representations, based on the results of these tests, have been plotted on plates 52 and 53.

TABLE C.—*U. S. S. Salem.*

Number of test.	1.		2.		3.		4.	
1. Date of test	July 17, 1909		July 18, 1909		July 19, 1909		July 20, 1909	
2. Duration of test ... hours		12		10		8		6
3. Kind of fuel	Georges Creek.		Georges Creek.		Georges Creek.		Georges Creek.	
4. Kind of start	Continuous fires.		Continuous fires.		Continuous fires.		Continuous fires.	
5. State of weather	Fair.		Fair.		Fair.		Fair.	
AVERAGE PRESSURES.								
6. Barometer ... inches		29.90		29.88		29.76		30.24
7. Steam pressure by gage ... pounds		242		221		245		247
8. Force of draft at base of pipe ... inches of water		−0.4		−0.6		−0.7		−0.7
9. Force of draft in furnace ... do	0.16	0.17	0.81	0.78	1.59	1.47	2.67	2.66
10. Force of draft in fireroom ... do	0.5	0.5	1.25	1.25	2.5	2.5	4.0	4.0
11. Revolutions of blower		195		294		405		512
AVERAGE TEMPERATURES.								
12. External air ... °F		77		79		72		66
13. Fireroom ... do		115		110		101		95
14. Steam ... do		403		396		405		405
15. Feed water entering heater ... do	Not in use.		Not in use.		Not in use.		Not in use.	
16. Feed water entering economizer ... do	None.		None.		None.		None.	
17. Feed water entering boiler ... do		127		90		117		111
18. Air entering ash pit ... do		115		110		101		95
19. Escaping gases from boiler ... do		476		558		632		684
20. Escaping gases from economizer ... do	None.		None.		None.		None.	
FUEL.								
21. Kind of	Georges Creek.		Georges Creek.		Georges Creek.		Georges Creek.	
22. Weight of wood used in lighting fires ... pounds	....		....		....		....	
23. Weight of coal as fired ... do		23,800		35,000		40,650		42,600
24. Moisture in coal ... per cent		1.44		Ave. 1.11		0.78		Ave. 1.11
25. Weight of dry coal consumed ... do		23,458		34,612		40,333		42,127
26. Weight of ash by analysis ... pounds		1,560		2,721		3,001		3,016
27. Weight of combustible consumed ... do		22,218		32,247		37,625		39,552
28. Per cent of ash by analysis in dry coal		6.65		7.86		7.44		7.16
FUEL PER HOUR.								
29. Coal consumed per hour ... pounds		1.983		3,500		5,081		7,100
30. Dry coal consumed per hour ... do		1,955		3,461		5,042		7,021
31. Combustible consumed per hour ... do		1,851		3,225		4,703		6,592
32. Coal consumed per hour per square foot G. S. ... do		17.09		30.17		43.80		61.20
33. Dry coal consumed per hour per square foot G. S. ... pounds		16.85		29.84		43.47		60.53
34. Combustible consumed per hour per square foot G. S. ... pounds		15.96		27.80		40.54		56.83
35. Coal per hour per square foot H. S. ... do		0.313		0.552		0.802		1.121
36. Dry coal per hour per square foot H. S. ... do		0.308		0.546		0.796		1.109
37. Combustible per hour per square foot H. S. ... do		0.292		0.509		0.743		1.041

TABLE C.—*U. S. S. Salem*—Continued.

Number of test.	1.	2.	3.	4.
QUALITY OF STEAM.				
38. Per cent of moisture in steam	0.241	0.646	0.44	0.671
39. Degrees of superheating	None.	None.	None.	None.
40. Quality of steam (dry steam=100)	99.759	99.354	99.56	99.328
WATER.				
41. Total weight of water fed to boiler [a] ... pounds..	230,683	298,836	347,349	329,883
42. Water actually evaporated, corrected for quality of steam (40 by 41) ... pounds..	230,127	296,905	345,821	327,666
43. Factor of evaporation	1.15	1.186	1.161	1.167
44. Equivalent water evaporated into dry steam from and at 212° (42 by 43) ... pounds..	264,646	352,129	401,152	382,386
WATER PER HOUR.				
45. Water evaporated per hour, corrected for quality of steam ... pounds..	19,177	29,690	43,228	54,611
46. Equivalent evaporation from and at 212°	22,054	35,213	50,144	63,731
47. Equivalent evaporation from and at 212° per square foot G. S. ... pounds..	190	303	432	549
48. Same per square foot of heating surface ... do....	3.48	5.56	7.92	10.06
ECONOMIC RESULTS.				
49. Water apparently evaporated under actual conditions per pound of coal as fired (41÷23) ... pounds..	9.69	8.54	8.54	7.71
50. Apparent equivalent evaporation from and at 212° per pound of coal (including moisture) 44÷23, pounds	11.12	10.06	9.84	8.94
51. Equivalent evaporation from and at 212° per pound of dry coal (44÷25) ... pounds..	11.28	10.17	9.95	9.07
52. Equivalent evaporation from and at 212° per pound of combustible (44÷27) ... pounds..	11.91	10.92	10.66	9.67
EFFICIENCY.				
53. Efficiency of boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of one pound of combustible ... per cent..	75.49	69.48	66.65	60.96
54. Efficiency of boiler, including grate; heat absorbed by the boiler per pound of dry coal, divided by the heat value of one pound of dry coal	....	....	....	....
REMARKS AND OBSERVATIONS.				
55. Principal data taken every	Hour.	Hour.	Hour.	Hour.
56. Percentage of smoke as observed		No estimate made.		
57. Average thickness of fires	4″ to 5″	5″ to 6″	6″ to 7″	8″ to 9″
58. Efficiency of firemen; expert, average, or poor		Good average fireman.		

[a] Corrected for inequality of water level and steam pressure at beginning and end of test.

TABLE C.—*U. S. S. Salem*—Continued.

FUEL AND GAS ANALYSES.

PROXIMATE ANALYSIS OF FUEL.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Fixed carbon	73.76	71.82	72.68	72.85
Volatile matter	18.93	18.67	18.61	18.90
Moisture	0.82	1.38	1.19	1.05
Ash	6.49	8.13	7.52	7.20
Total	100.00	100.00	100.00	100.00
Sulphur separately determined	0.927	1.04	0.95	0.93

ULTIMATE ANALYSIS OF DRY FUEL.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Carbon (C)	83.10	81.40	81.69	81.85
Hydrogen (H)	4.57	4.62	4.63	4.54
Oxygen (O)	3.38	3.85	4.07	4.22
Nitrogen (N)	1.37	1.22	1.22	1.29
Sulphur (S)	0.93	1.05	0.95	0.94
Ash	6.65	7.86	7.44	7.16
Total	100.00	100.00	100.00	100.00
Moisture in sample of fuel as received	0.82	1.38	1.19	1.05

CALORIFIC VALUE OF FUEL.

Number of test.	1.	2.	3.	4.
	B. T. U.	*B. T. U.*	*B. T. U.*	*B. T. U.*
Calorific value by calorimeter, per pound of dry coal	14,430	14,135	14,419	14,370
Calorific value by calorimeter, per pound of combustible	15,455	15,339	15,589	15,477
Calorific value by analysis, per pound of dry coal	14,705	14,451	14,482	14,436
Calorific value by analysis, per pound of combustible	15,751	15,682	15,645	15,548

ANALYSES OF DRY GASES.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Carbon dioxide (CO_2)	7.16	8.30	9.83	10.10
Oxygen (O)	9.06	11.03	10.60	9.15
Carbon monoxide (CO)	0.88	0.28	1.24	1.23
Nitrogen (N) (by difference)	82.90	80.39	78.33	79.52
Total	100.00	100.00	100.00	100.00

TABLE D.—*U. S. S. Chester.*

Number of test.	1.	2.	3.	4.
1. Date of test	Feb. 2, 1909.	Feb. 3, 1909.	Feb. 4, 1909.	Feb. 5, 1909.
2. Duration of test ... hours	10	8	6	12
3. Kind of fuel	Eureka.	Eureka.	Eureka.	Eureka.
4. Kind of start	Continuous fires.	Continuous fires.	Continuous fires.	Continuous fires.
5. State of weather	Fair.	Fair.	Fair.	Fair.
AVERAGE PRESSURES.				
6. Barometer ... inches	....	....	....	....
7. Steam pressure by gage ... pounds	258.5	257	258	252
8. Force of draft at base of pipe ... inches of water	−0.4	−0.6	−0.7	−0.4
9. Force of draft in furnace ... do	0.0 0.02	0.42 0.45	1.1 1.22	−0.2 −0.2
10. Force of draft in fireroom ... do	0.5 0.5	1.25 1.25	2.5 2.5	Natural draft.
11. Revolutions of blower		Not taken.		
AVERAGE TEMPERATURES.				
12. External air ... °F		Do.		
13. Fireroom ... do	77	65	62	88.7
14. Steam ... do	408.5	408	408.5	406.5
15. Feed water entering heater ... do	103	87	86	103
16. Feed water entering economizer ... do	None.	None.	None.	None.
17. Feed water entering boiler ... do	143	115	98	215
18. Air entering ash pit ... do	77	65	62	88.7
19. Escaping gases from boiler ... do	655	722	973	602
20. Escaping gases from economizer ... do	None.	None.	None.	None.
FUEL.				
21. Kind of	Eureka.	Eureka.	Eureka.	Eureka.
22. Weight of wood used in lighting fires ... pounds	None.	None.	None.	None.
23. Weight of coal as fired ... do	33,200	36,450	38,550	25,875
24. Moisture in coal ... per cent	1.5	2.75	2.0	Ave. 1.92
25. Weight of dry coal consumed ... pounds	32,702	35,448	37,779	25,379
26. Weight of ash by analysis ... do	2.093	2,091	2,875	1,716
27. Weight of combustible consumed ... do	31,080	34,300	35,620	24,125
28. Per cent of ash by analysis in dry coal	6.40	5.90	7.61	6.76
FUEL PER HOUR.				
29. Coal consumed per hour ... pounds	3,320	4,556	6,425	2,156
30. Dry coal consumed per hour ... do	3,270	4,430	6,296	2,115
31. Combustible consumed per hour ... do	3,108	4,287	5,937	2,010
32. Coal consumed per hour per square foot G. S. ... do	28.62	39.28	55.39	18.58
33. Dry coal consumed per hour per square foot G. S., pounds	28.19	38.19	54.28	18.23
34. Combustible consumed per hour per square foot G. S. ... pounds	26.79	36.96	51.18	17.33
35. Coal per hour per square foot H. S ... do	0.622	0.853	1.203	0.403
36. Dry coal per hour per square foot H. S ... do	0.612	0.829	1.179	0.396
37. Combustible per hour per square foot H. S ... do	0.582	0.803	1.112	0.376
QUALITY OF STEAM.				
38. Per cent of moisture in steam	0.173	0.31	0.401	0.143
39. Degrees of superheating	None.	None.	None.	None.
40. Quality of steam (dry steam=100)	99.827	99.69	99.599	99.862

TABLE D.—*U. S. S. Chester*—Continued.

Number of test.	1.	2.	3.	4.
WATER.				
41. Total weight of water fed to boiler [a] ....pounds..	273,896	287,031	265,766	237,759
42. Water actually evaporated, corrected for quality of steam (40 by 41)........................pounds..	273,420	286,141	264,700	237,430
43. Factor of evaporation..............................	1.134	1.163	1.180	1.058
44. Equivalent water evaporated into dry steam from and at 212° (42 by 43)...................pounds..	309,605	332,381	312,160	250,963
WATER PER HOUR.				
45. Water evaporated per hour, corrected for quality of steam.................................pounds..	27,342	35,769	44,117	19,786
46. Equivalent evaporation from and at 212°............	30,960	41,548	52,026	20,914
47. Equivalent evaporation from and at 212° per square foot G. S...............................pounds..	267	358	448	180
48. Same per square foot of heating surface......do....	5.80	7.78	9.74	3.91
ECONOMIC RESULTS.				
49. Water apparently evaporated under actual conditions per pound of coal as fired (41÷23) ..pounds..	8.25	7.88	6.89	9.19
50. Apparent equivalent evaporation from and at 212° per pound of coal (including moisture) 44÷23 ....pounds..	9.33	9.12	8.10	9.70
51. Equivalent evaporation from and at 212° per pound of dry coal (44÷25).....................pounds..	9.46	9.38	8.26	9.88
52. Equivalent evaporation from and at 212° per pound of combustible (44÷27)................pounds..	9.96	9.69	8.76	10.40
EFFICIENCY.				
53. Efficiency of boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of 1 pound of combustible............per cent..	62.46	59.47	55.84	65.58
54. Efficiency of boiler, including grate; heat absorbed by the boiler per pound of dry coal, divided by the heat value of 1 pound of dry coal.	No analysis was made of the ashes nor of the cinder blown out of the stack.			
REMARKS AND OBSERVATIONS.				
55. Principal data taken every........................	½ hour.	½ hour.	½ hour.	½ hour.
56. Percentage of smoke as observed..................		No estimate made.		
57. Average thickness of fires..........................	5″ to 6″	6″ to 7″	8″ to 9″	4″ to 5″
58. Efficiency of firemen; expert, average, or poor.....		Good average firemen.		

[a] Corrected for inequality of water level and steam pressure at beginning and end of test.

TABLE D.—*U. S. S. Chester*—Continued.

FUEL AND GAS ANALYSES.

PROXIMATE ANALYSIS OF FUEL.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Fixed carbon	76.45	76.84	75.51	76.61
Volatile matter	16.17	15.75	15.51	15.36
Moisture	.52	.97	.78	.89
Ash	6.87	6.44	8.20	7.14
Total	100.00	100.00	100.00	100.00
Sulphur separately determined	1.36	1.53	1.70	1.14

ULTIMATE ANALYSIS OF DRY FUEL.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Moisture in sample of fuel as received	0.52	0.97	0.78	0.89
Carbon (C)	83.41	84.06	81.82	83.27
Hydrogen (H)	4.42	4.39	4.31	4.33
Oxygen (O)	3.30	3.03	3.49	3.44
Nitrogen (N)	1.10	1.08	1.06	1.05
Sulphur (S)	1.37	1.54	1.71	1.15
Ash	6.40	5.90	7.61	6.76
Total	100.00	100.00	100.00	100.00
Moisture in sample of fuel as received	0.52	0.97	0.78	0.89

CALORIFIC VALUE OF FUEL.

Number of test.	1.	2.	3.	4.
	B. T. U.	*B. T. U.*	*B. T. U.*	*B. T. U.*
Calorific value by calorimeter, per pound of dry coal	14,629	14,697	14,320	14,548
Calorific value by calorimeter, per pound of combustible	15,627	15,617	15,498	15,600
Calorific value by analysis, per pound of dry coal	14,673	14,744	14,370	14,576
Calorific value by analysis, per pound of combustible	15,674	15,668	15,551	15,630

ANALYSES OF DRY GASES.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Carbon dioxide (CO_2)	6.33	7.16	9.65	9.14
Oxygen (O)	15.04	11.60	9.43	9.81
Carbon monoxide (CO)	0.34	0.76	0.23	0.53
Nitrogen (N) (by difference)	78.29	80.48	80.69	80.52
Total	100.00	100.00	100.00	100.00

III. STANDARDIZATION RUNS.

DATES OF DRY DOCKING, ETC.

Speed performances of vessels are considerably and variably affected by the state of wind and sea, and especially by condition of underwater hull as to cleanliness. Reduction in speed invariably follows sea growth accumulations on ships' hulls, being most rapid and marked with unsheathed vessels of steel construction. In order to minimize the uncertain and varying influence on speed, attributable to this, during comparative sea trials of the Scouts, it was the aim to run all such tests with ships' bottoms clean. This plan, however essential to accurate comparison, proved a most difficult one to execute with entire satisfaction, and, no doubt seemingly small, inexplainable discrepancies of speeds and power on some of the various trials are in a large measure due to this cause.

As a preliminary to standardization runs, the three vessels were dry docked and the bottoms cleaned and painted with the same variety of (McInnes) paint. The *Birmingham* was in dock September 17–24, the *Salem* (first standardization) from September 25 to October 1, and the *Chester*, September 14 to 16, all dates mentioned being in the year 1908. The *Birmingham* and *Salem* received a touching-up coat of anticorrosive, and later a complete coat of anticorrosive and antifouling paint. As the *Chester* had only been out of dry dock about a month, it was only considered necessary, when again docked at the date noted, to apply a touching-up coat of anticorrosive and a complete coat of antifouling paint.

After coming out of dry dock and prior to standardization runs on December 14, 1908 (pl. 85), the *Birmingham* did little sea cruising, covering less than 700 miles, the intervening time being spent alongside the wharf at the Boston Navy-Yard, re-expanding boiler tubes, and completely replacing tubes in boilers A and B.

The *Salem*, in the meantime, was ordered on a preliminary shaking-down cruise to the West Indies, during which time approximately 5,000 miles was covered. The ship left Boston on October 17, 1908, and returned to that port on November 19, 1908, when it was decided to fit new propellers, the vessel being placed in dry dock for that purpose from November 30 to December 3, 1908, and at that time a touching-up coat of antifouling paint was applied to the bottom. The first standardization runs (pl. 86) were made on December 15, 1908. This vessel, however, was re-standardized (pl. 87) while the second series of steam-consumption tests were in progress (Aug. 11, 1909), but just prior to this (July 2 to 8, 1909) the ship was dry docked and a complete coat of anticorrosive and antifouling paint was applied to the bottom. Comparison of the results of the two standardizations, the displacement being practically the same for each, shows material differences in horsepower and revolutions required for a given speed, and this is referred to more at length under steam-consumption trials.

The *Chester*, after dry docking, remained at the Boston Navy-Yard until October 14, 1908, and then made a cruise in northern waters of about 1,450 miles, arriving finally at Rockland, Me., on November 3, 1908. Standardization of this vessel (pl. 88) extended over a period of three days (Nov. 5, 6 and 7, 1908) and involved running over the meas-

ured-mile course, using all three—4, 5, and 6 turbine—combinations of main propelling machinery. Referring to plate 88, it will be seen that 12 runs each were made with the 5 and 6 turbine combinations and 17 runs with the 4-turbine combination, all of which were with exhaust steam from the various auxiliaries in operation directed into the feed-water heaters, boiler steam only being admitted to the turbines. For the purpose of ascertaining if a given mean (all shafts) number of revolutions resulting by admitting exhaust steam from the auxiliaries into the turbines would cause a difference from the speed corresponding to the same mean number of revolutions, using boiler steam only, 3 experimental runs (6-turbine combination) were made with the auxiliary exhaust open to the second stage of the main H. P. turbines. This resulted in a mean average speed of 14.57 and corresponding mean average revolutions of 301.5, with an aggregate S. H. P. of 2,570. Three additional experimental runs were carried out (6-turbine combination in use) with the auxiliary exhaust steam directed into the L. P. turbines, which gave a mean average speed of 15.28 and corresponding mean average revolutions of 312.8, with a total S. H. P. of 3,100. Both series of experimental runs are plotted on plate 88. But one speed-revolution curve, however, is laid down for all turbine combinations up to 19.5 knots. Above this speed, and for the 4 and 5 combinations, separate speed-revolution curves are drawn.

All vessels were again docked in 1909, just prior to coal-consumption tests on dates as follows: The *Birmingham*, March 12 to 16; the *Salem*, March 12 to 14; the *Chester*, March 8 to 12. The bottoms of all vessels received a complete coat each of anticorrosive and anti-fouling, and the *Birmingham*, in addition, a preliminary touching-up coat of anticorrosive, paint.

It may be pertinent, on account of the importance which foulness of bottom bears upon speed, power, etc., to recapitulate here the number of days intervening between the date of coming out of dock and commencement of the various trials.

	Days out of dock.		
	Birmingham.	Salem.	Chester.
Before standardization trials	80	12	49
Before commencement of main steam consumption tests	104	27	61
Before first coal-consumption test	4	6	8

In the case of the *Salem*, for the second series of tests, standardization took place thirty-three days and steam consumption tests began twenty-nine days after the vessel left dry dock.

IV. STEAM CONSUMPTION TRIALS OF MAIN PROPELLING AND AUXILIARY MACHINERY.

Of the various trials carried out under the Board's supervision, those of this group, comprising tests to fix steam consumption of the propelling machinery of the three vessels at different sea speeds, are considered to be of the most importance. The object was to

establish the comparative steam expenditures of the three types of main propelling installations, progressively, from about 10 knots to maximum speed, and, incidentally, this involved determination of the steam used collectively on each trial by the auxiliaries in operation. It will be apparent that in running these trials, it was necessary to keep in view, and, as far as possible, guard against any features which might have an indeterminate influence on speed, such as the following: (1) Unlike conditions of underwater hull as to cleanliness; (2) variable weather and sea conditions; (3) dissimilarity in displacement.

All vessels were fitted with separate and similar water measuring apparatus, and the series of tests on each were undertaken at different times, as the ships could be spared for the purpose. When once started, however, the entire series was finished before assignment of the vessel to duty elsewhere, and for convenience in coaling as the trials progressed, the coaling plant at Bradford, R. I., was selected as an accessible base from which to operate. For the second series of tests on the *Salem*, the coaling station at Frenchmans Bay was chosen as a base.

It was the aim to have the bottoms of all vessels free from marine growth, and as far as could be judged from the condition of that part of the underwater body visible at the water-line there was no radical difference in this respect when the trials were run. The time interval between coming out of dry dock and commencement of these trials, however, was not, as has been previously pointed out, the same for any of the three vessels.

Tests were run during favorable weather only; that is to say, under conditions of smooth sea and comparatively light winds, and, as far as practicable, in day time, the ship always coming to anchor about nightfall. Frequent delays occurred by reason of unfavorable weather, and often, on slow speed runs, advantage was taken of the protected waters of Long Island Sound in order to escape unsuitable weather conditions prevailing in the open sea.

An effort was made to maintain uniformly a standard mean displacement of 4,000 tons on all trials, but evidently this could be carried out practically within reasonably narrow limits only. The draft was adjusted by water ballast (using trimming tanks and double bottom compartments) at the beginning of each day, when tests were to be made, so that the resulting displacement would be slightly over the figure stated, due consideration being given to the amount of coal likely to be consumed. The displacement for each trial run during the day was readily figured from the draft on coming to anchor for the night. As the ship's displacement gradually decreased, due to daily fuel consumption, a condition was finally reached, after completion of a number of tests, where it was impracticable to arrange for the standard displacement by water ballast alone; at such times tests were temporarily discontinued until the ship could be again coaled.

A brief description has been given of the water measuring appliances and it will only be necessary to set forth here more in detail the method followed in its operation. Reference has been made to lift pumps which were installed as part of the test outfit. Three such pumps were provided on each vessel, one in connection with each main feed tank, and one for the auxiliary measuring tank. The

sole function of the main lift pumps was to draw water from the feed tanks and elevate it to the measuring tanks located on the spar deck. From these latter, after measurement, the water was dropped by gravity into distributing tanks below, from which, through proper connections to the feed pumps, the boilers were supplied with feed water. The auxiliary lift pump handled all water, after measurement from the auxiliary tank, and delivered it to the forward distributing tank; water of condensation from the auxiliaries in operation was discharged into the auxiliary measuring tank by the air pump of the auxiliary condenser.

It was the usual practice, after the propelling machinery had been regulated to the number of revolutions desired (as determined from the standardization speed-revolution curve) and with the water of condensation passing through both main and auxiliary measuring apparatus, to allow about one-half hour to intervene before beginning to record data, in order to guard against irregularities of operation and for the further purpose of permitting the machinery to assume normal working conditions. Eight observers were stationed as follows: one at each main measuring tank; one at the auxiliary measuring tank; two in each engine room to record data of counters, gages, revolutions, etc., of auxiliary machinery, and height of water in feed tanks; one to record data of auxiliaries outside of the engine room spaces. In addition to the above, a requisite number of observers were assigned to secure data from which the power of the propelling machinery could be calculated.

A stand-by signal was given one minute before each test began, generally by engine-room gongs; also at the instant of beginning each test, and thereafter at precisely twenty-minute intervals as the test progressed. At the moment of starting, measuring-tank observers shifted goose-neck connections, and thus directed the water of condensation into the empty compartment of the measuring tanks, the valves at the bottoms controlling the outlets having been previously closed. As each measuring-tank compartment filled, the remaining one was emptied and prepared for the reception of water, so that both compartments were used alternately as the test proceeded. A record was kept of the elapsed time in filling each compartment, as well as of the temperature of the water.

In the engine room the height of water in each feed tank at all time intervals during the tests was recorded, a graduated scale being attached to facilitate making such entries. These tanks, moreover, had been calibrated, and although the aim in all tests was to maintain the water delivered into them at a constant height, in event of departure from this, the exact weight of exhaust steam condensed by each main condenser could be figured at any time by applying a proper correction. The feed tanks were also connected by an equalizing pipe, making it possible to use but one main measuring tank and its lift pump for both engine rooms; and it may be noted in passing that during the first tests on the *Chester* the forward lift pump gave considerable trouble from unsatisfactory working, unavoidably necessitating on such tests connection of the two feed tanks through the equalizing pipe. Ordinarily, however, the tanks were disconnected, as this method furnished the means of separately determining the weight of exhaust steam entering each main condenser.

Steam-consumption tests may be classed under two general headings, as follows: First, those in which none of the auxiliary exhaust steam was used in the main propelling machinery to assist propulsion; and, second, those in which all available auxiliary exhaust, not utilized in the feed heaters, was conveyed to the propelling machinery to assist propulsion. Otherwise stated, the two classes comprise tests, (1) in which exhaust steam from the auxiliaries was led into the feed-water heaters, afterwards being delivered to and condensed in the auxiliary condenser, and (2) tests in which the auxiliary exhaust line was open, not alone to the feed heaters, but as well to L. P. engine receivers (*Birmingham*) or some stage (*Salem* and *Chester*) of main turbines. In tests of the latter character only condensed exhaust from dynamos, trap drains, etc., delivered into the auxiliary measuring tank, the regulating valve in auxiliary exhaust line at the auxiliary condenser being tightly closed to prevent entry therein of exhaust steam. It will be evident that water of condensation collected and measured in the main tanks in such tests included steam used by the main propelling machinery, and in addition part of that used by the auxiliaries. A pressure above the atmosphere was always maintained in the auxiliary exhaust line, which was effected by a regulating valve on the auxiliary condenser in tests of the class first mentioned, and by manipulation of proper valves attached to propelling machinery on other tests. It should be noted that the larger the percentage of auxiliary exhaust steam delivered to the main engines or turbines the greater the speed and power developed, the highest speed and power on a given steam consumption being obtained when the feed heaters are not in use. The apparent economy, as measured in total steam per knot and per horsepower, would then reach a maximum. This, however, does not represent the point of maximum economy when the speed and power are based upon coal consumption, for the use of sufficient auxiliary exhaust in the feed heaters to give the feed-water the highest temperature increases the amount of steam evaporated in the boilers when the same amount of coal is being consumed, so that the increased output more than counterbalances the apparent loss due to the use of exhaust steam in feed-water heaters. This fact should always be taken into consideration when the performance of any vessel is given in terms of total water per knot or per horsepower, otherwise misleading inferences may be drawn. It may be further pointed out in connection with steam-consumption trials that close regulation of auxiliaries to the requirements (which to a large extent are dependent upon speed of vessel) was not attempted. In this respect data of coal-consumption trials furnish a safer guide as to the least quantity of steam necessary for operation of the auxiliaries, since on all ships during these trials every effort was made to reduce this expenditure to a minimum.

Power of the main propelling machinery was calculated, for the *Birmingham*, from indicator cards. Torsion meters, of the Denny-Johnson type, were fitted on the *Salem* and *Chester* to obtain shaft horsepower, determinations with these instruments being made by comparison of the torsion on a length of turbine shafting, which before installation had been carefully calibrated, and torsion readings on the same length of shaft during the trials. In the latter, the torsion meters measured the torsion by means of magnets, electrical resistance coils, etc., readings depending for accuracy upon adjustments

for sound disappearance. Particular care was exercised, at all times, to secure accurate torsion meter readings, but the type mentioned as installed on these vessels is believed to be too delicate, both in construction and theory of operation, for satisfactory and reliable results on shipboard. Moreover, at low powers, because graduations for sound disappearance were so widely apart, a source of considerable possible error existed, no matter how carefully readings were taken, as illustrated by the following: Scale graduation, between adjacent contact points on the instrument, progress by 0.02 beginning at 0. The smallest reading recorded on any test was 0.07, which lies between the contact points of 0.06 and 0.08, and which, therefore, can only be regarded as an approximation. Furthermore, this reading was subject to correction for determination of zero point of the instrument, which is established by steaming ahead, for example, on the port screws, while the starboard shafts revolve idly and presumably without torque. It will be evident that this latter reading can not be taken with any greater degree of accuracy than the former. At higher powers, the possibility of error is smaller in percentage of the total, and in addition, torsion-meter readings can be more readily noted, because at higher rotative speeds the sound is more distinguishable.

Since on various sea trials it was possible only to obtain shaft horsepower of the main propelling equipments of the *Salem* and *Chester*, while on the *Birmingham* torsion meters were not installed, the necessity of selecting either shaft or indicated horsepower as the unit for purposes of power comparison is apparent. Shaft horsepower has been chosen principally because power expended in propulsion on two of the three vessels was thus measured. Sufficient experimental data is not extant to fix accurately the existing ratio between shaft and indicated horsepower. Moreover, this ratio varies somewhat for the same engine, being greater at lower than at higher powers, and besides is not constant for engines of the same power if of different types. In practice, however, it is generally considered as constant for all usual ranges of power of the same engine. In calculations relating to these trials, a ratio of 0.94 has been assumed in converting indicated into shaft horsepower, but this should be regarded only as a close approximation. It may be further pointed out, in this connection, that no attempt has been made to convert indicated into shaft horsepower for the auxiliaries, since computations of this character introduce so many uncertainties, at best, as to render such calculations of doubtful value.

Curves of effective horsepower and speeds, as laid down from results of towing models (including the following appendages: Rudder, bilge keel, propeller shafts, and struts) of these vessels in the Washington model basin (4,000 tons displacement) are shown on plates 54, 55, and 56. Propulsive efficiency curve for the *Birmingham* has been laid down, but such curves for the other two vessels have not been drawn on account of variation and unquestionable inaccuracy of power recorded for the lower speeds.

Calorimeters of the Barrus throttling type were installed and readings therefrom recorded on all sea trials. These were located between the throttle and steam chest, as follows: One in each engine room of

the *Birmingham* and *Salem;* on the *Chester,* four in all, one each to H. P. cruising, I. P. cruising and main H. P. turbines.

Results of evaporative boiler tests heretofore given show practically no moisture in the steam, even for higher combustion rates. In calculations relating to steam consumption of main and auxiliary machinery, therefore, it has been assumed that dry steam was supplied at all times. In this connection it may be remarked that steam separators were fitted in each engine room, in the main steam line of the *Birmingham* and *Salem,* but no similar appliance was installed on the *Chester.* Through an oversight in design and construction, the interior diaphragm being improperly located, the forward separators on both ships were only partly efficient for the purpose intended, unless cross-connection pipe between port and starboard main steam lines in forward engine room was not used. As installed, and with valves open on the pipe mentioned, steam from the port boilers could pass directly to the forward propelling engine, the separator merely serving as a means of communication. The first series of steam consumption trials on the *Salem* were run with this cross-connection pipe in use, and on some tests violent fluctuations were noticeable of the lower calorimeter thermometer. This may be attributed to the fact that the forward turbine on all these tests, when making the same number of revolutions, used more steam than the after, and consequently the supply was no doubt drawn partially through this cross-connection pipe. In later trials, connection between port and starboard main steam lines was effected through piping in the firerooms only.

In carrying out steam consumption trials on main machinery the importance of obtaining uniformity in water rates, under like conditions of operation, was realized. The weight of condensed steam collected was checked at frequent intervals as the tests progressed, and each test continued until agreement of results, within reasonable limits, was shown. Almost all tests cover a time interval of at least two hours, and this is believed to be sufficiently long to secure reliable data when fluctuations in water rates are slight. Where, however, considerable variation occurred in this particular, due to unknown causes, tests were continued over a longer period. As will be observed from an examination of the tables and accompanying remarks, the data recorded in the first part of some of the tests were rejected as unreliable.

Charts of speed, as shown by Nicholson recording logs, for all steam consumption trials have been copied on sheets containing remarks, and which follow data tables of each test. It should be stated that accuracy of speed thus shown depends upon adjustment of instruments and connecting accessories and is influenced also by helm angle and by rolling, as well as pitching of the ship. The charts are believed to be of interest principally as graphically illustrating speed regularity during the various tests.

The following was the order in which main steam consumption tests took place: *Chester,* November 18 to December 5, 1908; *Salem* (first series), December 31, 1908, to January 3, 1909; *Birmingham,* January 7 to 12, 1909; *Salem* (second series), August 7 to 14, 1909.

A summary in tabular form of the important results on the three vessels is given below.

BIRMINGHAM.

[Tables 33 to 42.]

The auxiliary exhaust line of piping on this vessel is connected to the second receivers of main engines. Tests Nos. 2, 4, 7, and 9 were carried out with receiver valves slightly open, the object being to thus utilize any auxiliary exhaust steam not condensed in the feed-water heaters. It will be apparent that, by cutting out the latter, all exhaust from the auxiliaries becomes available to assist in propulsion, but necessarily this involves lowering the boiler feed-water temperature; and while in this way a less number of pounds of steam per unit of power developed unquestionably results no increase in economy, based upon coal consumed, is shown. Receiver valves were so regulated in these tests, therefore, as to maintain the feed-water temperature at about 150°.

Tests Nos. 1, 3, 5, 6, 8, and 10 were conducted with engine receivers valves closed and engine room auxiliary condenser in operation to care for such excess of auxiliary exhaust steam as was not condensed in the feed heaters.

Synopsis of results of all tests are shown in Table E.

TABLE E.—*Steam consumption tests.*

U. S. S. Birmingham.
(Tables 33 to 42.)

IV.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower. Machinery necessary to propulsion. Main engines. I. H. P.	Main engines. S. H. P.=.94×I. H. P.	Auxiliaries. I. H. P.	Total. I. H. P. (c+e).	Total. H. P. (d+e).	Condensed exhaust steam (pounds). Per hour. Main engines.	Per hour. All machinery necessary to propulsion.	Per hour. All purposes except lift pumps.	Per hour. All purposes.	Per hour. Per S. H. P. of main engines. $\frac{h}{d}$	Per hour. Per H. P. of all machinery necessary to propulsion. $\frac{k}{g}$	Per hour. Per I. H. P. of main engines. $\frac{h}{c}$	Per hour. Per I. H. P. for all machinery necessary to propulsion. $\frac{k}{f}$	Per knot. Used by main engines.	Per knot. Used for all purposes except lift pumps.	Per knot. Used for all purposes.	Auxiliaries exhaust into—
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u
1	10.03	1,050.2	987.2	52.1	1,102.3	1,039.3	21,988	26,689	33,797	34,069	22.3	25.7	20.9	24.2	2,192.2	3,369.5	3,396.7	Engine room auxiliary condenser.
2	10.65	1,236.9	1,162.7	48.0	1,284.9	1,210.7	....	....	33,292	33,561	......	......	......	......	....	3,126.0	3,151.3	Second receivers main engines.
5	14.95	2,948.1	2,771.2	99.1	3,047.2	2,870.3	49,755	58,729	64,889	65,412	18.0	20.5	16.9	19.3	3,328.1	4,340.4	4,375.4	Engine room auxiliary condenser.
3	a15.17	3,118.6	2,931.5	87.0	3,205.6	3,018.5	52,829	60,118	68,510	69,062	18.0	19.9	16.9	18.8	3,482.5	4,516.1	4,552.5	Do.
4	15.35	3,195.7	3,004.0	94.3	3,290.0	3,098.3	....	....	66,581	67,118	......	......	......	......	....	4,337.5	4,372.5	Second receivers main engines.
8	19.86	7,199.7	6,767.7	208.2	7,407.9	6,975.9	109,820	127,241	130,982	132,037	16.2	18.2	15.3	17.2	5,529.7	6,595.3	6,648.4	Engine room auxiliary condenser.
9	20.00	7,421.8	6,976.5	177.8	7,599.6	7,154.3	....	....	131,322	132,380	......	......	......	......	....	6,566.1	6,619.0	Second receivers main engines.

6	22.55	11,020.3	10,359.1	307.1	11,327.4	10,666.2	185,210	208,205	213,176	214,894	17.9	19.5	16.8	18.4	8,213.3	9,453.5	9,529.7	Engine room auxiliary condenser.
7	22.80	11,362.0	10,680.3	281.8	11,643.8	10,962.1	....	....	213,866	215,591	......	......	......	......	....	9,380.0	9,455.7	Second receivers main engines.
10	24.15	14,543.0	13,670.4	528.5	15,071.5	14,198.9	253,372	288,834	294,923	297,300	18.5	20.3	17.4	19.2	10,491.6	12,212.1	12,310.5	Engine room auxiliary condenser.

a Water collected in main measuring tanks unreliable. See remarks at end of test.

TABLE E.—*Steam consumption tests*—Continued.

IV.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressures (gage). Engine room. Forward.	Steam pressures (gage). Engine room. Aft.	Steam pressures (gage). Steam chest. Forward engine.	Steam pressures (gage). Steam chest. After engine.	Vacuum (inches of mercury). Forward engine room.	Vacuum (inches of mercury). After engine room.	Revolutions per minute.[a] Forward, engine starboard.	Revolutions per minute.[a] After engine, port.	Revolutions per minute.[a] Mean.	Indicated horsepower. Machinery necessary to propulsion. Main engines. Forward, starboard.	Main engines. After, port.	Main engines. Both.	Auxiliaries. Main air pumps.	Auxiliaries. Main circulating pumps.	Auxiliaries. Main feed pumps.	Auxiliaries. Forced draft blowers.	Auxiliaries. Total (r+s+t+u).	Grand total (q+v).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w
	H. m.																			
1	4 40	10.03	199.0	195.5	54.2	59.7	25.0	25.8	74.3	74.3	74.3	540.1	510.1	1,050.2	7.5	20.1	19.5	5.0	52.1	1,102.3
2	2 40	10.65	200.0	197.2	55.0	64.4	25.0	25.8	78.9	78.8	78.9	592.7	644.2	1,236.9	6.7	15.0	21.5	4.8	48.0	1,284.9
5	2 00	14.95	250.3	247.6	82.7	96.7	27.0	27.3	111.9	111.8	111.9	1,451.9	1,496.2	2,948.1	18.4	33.9	31.2	15.6	99.1	3,047.2
3	2 00	15.17	248.0	245.3	85.6	96.7	26.0	27.3	113.2	113.2	113.2	1,565.5	1,553.1	3,118.6	10.8	34.0	27.8	14.4	87.0	3,205.6
4	2 00	15.35	250.7	247.9	85.3	97.9	27.0	27.3	114.5	114.5	114.5	1,574.7	1,621.0	3,195.7	16.3	32.8	32.2	13.0	94.3	3,290.0
8	2 40	19.86	248.8	244.0	173.5	184.8	28.2	27.5	150.2	150.0	150.1	3,556.5	3,643.2	7,199.7	27.6	48.0	72.6	60.0	208.2	7,407.9
9	2 00	20.00	249.3	245.4	174.3	185.0	28.0	27.6	151.4	151.4	151.4	3,626.0	3,795.8	7,421.8	20.0	44.0	67.6	46.2	177.8	7,599.6
6	2 40	22.55	250.8	241.1	183.7	201.7	27.7	27.0	173.3	173.3	173.3	5,430.9	5,589.4	11,020.3	28.9	57.6	89.6	131.0	307.1	11,327.4
7	2 00	22.80	234.2	227.1	185.0	202.4	27.9	26.2	175.4	175.4	175.4	5,533.0	5,829.0	11,362.0	32.6	50.0	85.2	114.0	281.8	11,643.8
10	2 40	24.15	238.6	236.4	222.7	236.1	27.5	26.3	192.2	190.8	191.5	7,385.0	7,158.0	14,543.0	27.3	59.6	141.6	300.0	528.5	15,071.5

[a] As determined from counter readings at beginning and end of test.

SALEM.

[Tables 43 to 65.]

As previously pointed out, two series of steam-consumption trials were carried out on this vessel. The results of both series are given since the coal-consumption tests (V) and full-power run of twenty-four hours (VI) were made in the interval between the two. This made it necessary to run two standardization trials, the results of which, as to power for a given speed, differ materially, probably attributable to unlike weather conditions and different condition as to cleanliness of underwater hull when the trials were run. There is a likelihood, also, that torsion-meter readings were unreliable, or a further possibility of inaccurate determination of zero points of the instruments; investigation seems to show, however, slight probability of large differences in power being accounted for in this way. Both standardization curves are shown on plate 83, and in addition power points, as determined on all sea trials, are plotted on the same sheet. The latter points, with but one exception, it will be observed fall between the two standardization curves.

Upon completion of the full-power trial (VI), and because of the marked difference between the performance of the starboard and port turbines during both steam and coal consumption tests, the Fore River Ship Building Company requested, and was granted, permission to make an examination of the main turbines and undertake repairs found necessary. The starboard turbine had shown less efficiency than the port, inasmuch as more nozzles were required to be open on the former (with the same steam-chest pressure) to maintain the same revolutions of the two turbines. The vessel proceeded from New York Harbor (in which port she arrived after the trial last mentioned) to the works of the Fore River Company at Quincy, Mass. After removal of top castings and lifting of rotors a careful inspection of both turbines was made, the condition, in detail, being as set forth in a report forwarded to the Bureau of Steam Engineering by the inspector of machinery at the Fore River Works, and from which the following is taken:

Starboard or Forward Turbine.

It was found on examination that aside from the accidental damage which the starboard turbine had sustained and which accounted for the falling off in revolutions of this turbine compared with the port one, the general condition of both was equally bad, and that of the port turbine was perhaps worse than the starboard one. The accidental damage sustained by the starboard turbine was the almost complete closing up of the steam passages through the first row of buckets of the fifth stage wheel (see pl. 5) by some obstruction which was apparently picked up from the bottom of the casing and jammed between the moving buckets and the fifth stage nozzle blades. It gradually moved from one nozzle to the next, as the nozzle blades were bent down and the revolving bucket blades bent and hammered together, traveling in this manner over one quadrant of the nozzle circle to the top of the lower half casing. At this point it could advance no farther and remained lodged in the last nozzle opening until it had been battered and chafed a hole through the nozzle blade, when it probably dropped back into the fourth stage. By this time the obstruction, whatever it was, had probably been pounded and broken into small bits, which were carried on through the turbine, accounting in part for numerous minor injuries to the blading. No trace of it could be discovered in the turbine.

While the lodging of an obstruction between the fifth stage nozzles and the wheel buckets fully accounts for the injury sustained by the fifth stage wheel and nozzles, the causes of the bad condition of the turbine in other respects was to be found in the excessive working of the rotor and shaft fore and aft, made possible by the yielding of

the thrust foundations. In both turbines the rotor and shaft had worked fore and aft until the clearances in most of the stages between rotor buckets and intermediates had not only disappeared, but the rotor bucket bases and shrouds had rubbed and cut into the bases and shrouds of intermediates until the latter had, in many instances, been reduced in width from one-eighth to three-sixteenths of an inch. During this rubbing process the blades also had suffered injury, and in some rows the blades, particularly of the intermediates, had been dented and frayed, but on the whole the injury to the buckets was slight, owing to their springiness. It is to be noted, however, that while in the port or after turbine the movement of the rotor had been forward, yielding to the thrust of the screw and the steam thrust, which act in the same direction in this turbine, in the forward or starboard turbine the displacement of the rotor had been aft against the screw thrust, yielding to the abnormal steam thrust brought about by the closed buckets of the fifth stage wheel.

The main cause of the injuries discovered in both turbines, aside from the accidental damage to the nozzles and wheel of the fifth stage of the starboard turbine, and to the rotor buckets of the first stage wheel of the port turbine, was to be found in the lack of rigidity of the thrust foundations, which yield under stress. It was noticed in a number of the stages of both turbines that the buckets of rotor wheels had chafed on both the forward and the after edges of intermediates, showing that the entire clearance space, measuring about seven-sixteenths of an inch in the present condition of rotors and intermediates in stages in which chafing was observable on both edges of intermediates, had been bridged by the fore and aft movement. The movement fore and aft was probably not quite so great as this, as the chafing on opposite sides of intermediates possibly occurred at different times, the rotor being adjusted in the meantime, but such adjustment was certainly very small when compared with the total clearance. There is no doubt that excessive fore and aft spring did exist. On the run of the vessel from New York to the Fore River Engine Works a fore and aft movement of the forward end of thrust bearing of the port turbine had been observed, amounting to one-sixteenth of an inch at the level of the bearing pedestal flange and to three-sixteenths of an inch at about the height of shaft center. It speaks well for the design of the interior construction of the Curtis turbine that under such severe conditions the damage was comparatively easily remediable. The clearances became in many stages considerably in excess of the designed ones, but it is the opinion of the contractors that the efficiency will not be appreciably affected thereby.

The displacement fore and aft of the rotors and shafts caused trouble in the main bearings also, and in the forward rotor shaft stuffing box of the port turbine. The after main bearing of the starboard turbine heated excessively, owing to rubbing of the oil rings on the shaft against the end of the brass, the oil lip of the latter having been badly rubbed and cracked. Similar trouble also arose with the forward main bearing of the port turbine. Here also, the oil collars on the shaft had come in contact with the end of the brass, and in addition the shoulder of the shaft in the forward end of turbine casing had rubbed against the bottom of the stuffing box, resulting in excessive heating of the shaft and bearing. The bottom of the stuffing box was buckled and cracked, but not enough to make its removal a necessity. The heating of both the above main bearings took place during the first four hours of the twenty-four hour full-power run. Both were cooled with water hose, after which there was no further trouble.

Port or After Turbine.

Nearly all that is to be recorded of this turbine has been touched upon in speaking of the starboard turbine. As stated before, aside from the accidental injury to the fifth stage nozzles and wheel of the starboard turbine, the general condition of the port turbine was, if anything, the worse of the two, the injuries being due to the movement fore and aft of the rotor as described above. As before noted, the port turbine takes steam for going ahead at the after end, so that steam thrust and screw thrust act in the same direction, and any weakness in thrust bearing foundation will, therefore, be more seriously manifest. Besides the injuries due to the rotor displacement, this turbine had suffered accidental damages from a coarse-threaded five-eighths of an inch bolt, about 3 inches long, which was found, on lifting the top half casing, standing on its head in one of the second stage nozzle openings on the first stage side of diaphragm. The following injuries, which were confined to the first stage wheel were doubtless due to this bolt; the bolt itself showed that it had been heavily struck; one bucket and piece of rim or shroud in one segment of the first row of first stage wheel broken out; another segment in same row struck and rim cracked; rim badly dented in one place on fourth row.

The following is a report on the condition of the turbine parts in detail and of the measures adopted to repair defects, with comments. It should be noted that in the following the wearing away of intermediates and rotors spoken of has reference to wear

of bases and rims only, unless special mention is made of the blades. In most cases the latter escaped injury due to the rubbing, even where the solid metal of the bases and rims was much worn. Unless otherwise stated, bucket edges of rotors and intermediates reported as dented, bent, frayed, or nicked have been repaired by straightening and smoothing the edges:

Starboard Turbine.

FIRST STAGE AHEAD.

Nozzles.—Pieces broken out of the ends of five nozzle blades; one blade cracked at end and ready to break; two slightly bent. The defects in nozzle blades are not peculiar to those near the center of the group, which receive very much more wear than those nearer the ends, but occur irregularly. For example, in the nozzles of the starboard turbine four of the broken blades are found well toward the ends, one in the center. In the nozzle of the port turbine two of the broken blades are on the extreme end, the others near the middle. In view of this condition it is believed that the deterioration is not a matter of erosion, but of fatigue of the metal, which renders it brittle. The nozzle blades had been made unnecessarily thin, and in the new nozzles which were fitted they were left somewhat heavier.

Rotor buckets.—Worn down slightly on the after side, due to rubbing; a number of blades dented and a few chipped slightly.

Intermediates.—Bases and rims badly worn, due to rubbing of rotor, and edges of bucket blades frayed and bent. The rubbing occurred entirely on the forward side of intermediates and the wear amounts to from one-eighth to three-sixteenths of an inch, averaging five thirty-seconds of an inch. All intermediate segments were renewed.

SECOND STAGE.

Nozzles.—In good condition.

Rotors.—First two bucket rows slightly worn on after side, due to rubbing; a few blades in first row badly dented.

Intermediates.—Badly worn and bucket edges frayed; condition and amount of wear same as first stage intermediates; all intermediate segments were renewed.

THIRD STAGE.

Nozzles.—In good condition.

Rotors.—First two rows slightly worn on after side, due to rubbing; a number of blades in first row badly dented.

Intermediates.—Worn on forward side about one thirty-second of an inch and condition good; second row shows slight rubbing on after side also.

FOURTH STAGE.

Nozzles.—In good condition.

Rotors.—First two rows slightly rubbed on after side; a number of blades dented.

Intermediates.—General condition about the same as third stage intermediates; wear somewhat greater, scant one-sixteenth of an inch, but on forward side only.

FIFTH STAGE.

Nozzles.—In good condition, except those in lower inboard quadrant, blades of which were bent and worn by some obstruction not found. The injured sections, three in number, were renewed.

Rotors.—Badly worn by rubbing on after side; first row of buckets entirely closed by some obstruction which lodged in lower inboard nozzles; second and third rows of blades dented and cut, the second row more so than the third; the first and second rows were entirely renewed.

Intermediates.—Badly worn and bucket edges rubbed and frayed; entirely renewed.

SIXTH STAGE.

Nozzles.—In good condition.

Rotors.—Slight rubbing on after side; many small dents in blade edges.

Intermediates.—Both rows show decided wear, about one thirty-second of an inch on forward side, and second row also shows chafed spots on after side.

SEVENTH STAGE.

Nozzles.—In good condition.

Rotors.—Slight rubbing on after side; many blades dented, a few deeply.

Intermediates.—In good condition.

FIRST REVERSE STAGE.

Nozzles.—In good condition.

Rotors.—Slight rubbing on after side; many blades dented, some in first row badly.

Intermediates.—Very slight rubbing on forward side; a number of blades badly nicked.

SECOND REVERSE STAGE.

Nozzles.—In good condition.

Rotors.—Very slight rubbing on after side; a number of blades dented, some in first row badly.

Intermediates.—First row slightly rubbed on both sides, greatest wear about one-sixteenth of an inch.

PORT TURBINE.

FIRST STAGE AHEAD.

Nozzles.—Seven nozzle blades more or less broken out at ends; two bent and ready to slip off; nozzles renewed.

Rotors.—The first stage wheel suffered injury owing to a bolt having, in some way, fallen into the casing. A blade and piece of rim was broken from one segment in the first row, and another segment had been struck and the rim cracked and loosened. The rim on one segment in the fourth row was badly dented and blades bent. The three damaged segments were renewed. First three rows show excessive rubbing and there are many nicked and dented blades.

Intermediates.—Bases and rims worn down badly by rubbing on after sides, from one-eighth to three-sixteenths of an inch, and blade edges also badly rubbed and frayed. Decided wear due to rubbing, amounting to one thirty-second of an inch as a maximum, also on the forward sides. All intermediate segments were renewed.

SECOND STAGE.

Nozzles.—In good condition.

Rotors.—Show slight rubbing on forward sides; many blades slightly nicked.

Intermediates.—Condition about the same as first stage intermediates; all segments renewed.

THIRD STAGE.

Nozzles.—In good condition.

Rotors.—Show rubbing on forward sides.

Intermediates.—Decided wear due to rubbing on after sides, about three thirty-seconds of an inch, also blade edges frayed and nicked. Traces of wear also on forward sides. These intermediates were not renewed.

FOURTH STAGE.

Nozzles.—In good condition.

Rotors.—Show rubbing on forward side, otherwise in good condition.

Intermediates.—Condition somewhat worse than that of third-stage intermediates, blade edges having suffered more; wear about one-eighth of an inch. These intermediates were not renewed.

FIFTH STAGE.

Nozzles.—In good condition.

Rotors.—Very slight rubbing on forward side; condition good.

Intermediates.—Slight rubbing on after side, scant one thirty-second of an inch. A few buckets very slightly dented.

SIXTH STAGE.

Nozzles.—In good condition.

Rotors.—Very slight rubbing on forward side.

Intermediates.—Slight rubbing on after side.

SEVENTH STAGE.

Nozzles.—In good condition.

Rotors.—Very slight rubbing.

Intermediates.—Very slight rubbing on after side; some blade edges dented.

FIRST REVERSE STAGE.

Nozzles.—In good condition.
Rotors.—In good condition; first row shows slight rubbing against nozzle plate.
Intermediates.—No rubbing; blade edges somewhat frayed and nicked.

SECOND REVERSE STAGE.

Nozzles.—Nozzle plate cracked in a number of places by rubbing of rotor.
Rotors.—First row badly worn because of rubbing against nozzle plate; second and third rows show rubbing and blades dented in many places. The first row was renewed.
Intermediates.—Slight wear on after side; blades nicked in many places.

To prevent thrust bearings from swaying fore and aft under the stress, braces were run from the thrust-bearing pedestals, diagonally to the supporting brackets on the first stage ahead, lower casing section. For the forward turbine, these braces are in tension when going ahead, while for the after turbine, they are in compression.

All movable valves (of the slide type) between stages, which were fitted over diaphragm nozzle openings, were removed at this overhauling. These valves were originally installed with the view of controlling to a limited extent the pressure and consequently the flow of steam in passing through the turbines. They were spaced at intervals circumferentially, and arranged to be hand operated through a stem extending to the outside casing. As designed, it was believed that by regulation of the steam flow through the various stages at low powers, by means of these valves, steam economy might thus be effected. Practically, however, as demonstrated during the first series of tests (Nos. 3 and 5) there appeared to be no material difference in steam economy, no matter whether these valves were closed or open.

Upon completion of repairs to the *Salem's* turbines, the vessel was ordered on a cruise to the Canary Islands, later returning to the Boston navy-yard for dry docking and reinstallation of water-measuring apparatus before commencement of the second series of steam-consumption trials. Data of both series are given in Tables 43 to 65, and a summary of results in Tables F and G below.

TABLE F.—*Steam consumption tests (first series).*

U. S. S. Salem.
(Tables 43-51.)

IV.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

		Horsepower.			Condensed exhaust steam (pounds).									
		Machinery necessary to propulsion.			Per hour.						Per knot.			
Number of test.	Speed per hour in knots.	S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines. $\frac{f}{c}$	Per H. P. of all machinery necessary to propulsion. $\frac{g}{e}$	Used by main turbines. $\frac{f}{b}$	Used for all purposes except lift pumps. $\frac{h}{b}$	Used for all purposes. $\frac{k}{b}$	Auxiliaries exhaust into—
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
2	9.8	619	107.1	726.1	32,519	40,296	46,027	46,398	52.5	55.5	3,318.3	4,696.6	4,734.5	Engine room auxiliary condenser.
3	10.1	654	106.0	760.0	33,278	40,896	45,731	46,100	50.9	53.8	3,294.8	4,527.8	4,564.3	Do.
4	10.18	711	134.2	845.2	....	....	46,644	47,020	....	....	....	4,581.9	4,618.8	Third stage forward turbine.
5	10.2	681	124.0	805.0	33,152	41,539	46,225	46,598	48.7	51.6	3,250.2	4,531.7	4,568.4	Engine room auxiliary condenser.
1	10.33	779	....	....	....	....	....	....	....	....	....	....	....	Do.
6	15.25	2,646	147.3	2,793.3	71,671	82,269	85,945	86,641	27.1	29.4	4,700.0	5,635.7	5,681.4	Do.
7	20.16	7,078	229.8	7,307.8	137,182	152,810	156,259	157,519	19.4	20.9	6,804.6	7,751.0	7,813.4	Do.
9	22.55	10,167	513.9	10,680.9	191,306	220,039	223,028	224,826	18.8	20.6	8,483.6	9,447.0	9,970.1	Do.
8	24.37	14,767	659.4	15,426.4	257,016	292,656	297,176	299,572	17.4	19.0	10,546.4	12,194.3	12,292.6	Do.

TABLE F.—*Steam consumption tests (first series)*—Continued.

IV.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage).				Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.										
			Engine room.		Steam chest.							Machinery necessary to propulsion.										Grand total (q+x).
												S. H. P. main turbines.			I. H. P. auxiliaries.							
			Forward.	After.	Forward.	After.	Forward engine room.	After engine room.	Forward starboard.	After port.	Mean.	Forward starboard.	After port.	Both.	Wet vacuum pumps.	Dry vacuum pumps.	Main circulating pumps.[a]	Main feed pumps.	Forced-draft blowers.	Oil pumps.	Total (r+s+t+u+v+w).	
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y
	H. m.																					
2	2 00	9.8	218.3	221.8	165.0	195.3	28.4	28.6	133.9	139.4	136.65	296	323	619	17.1	34.2	20.0	21.0	14.0	.85	107.1	726.1
3	2 00	10.1	223.4	222.1	185.0	192.7	28.4	28.6	141.1	140.5	140.8	329	325	654	17.4	34.5	19.6	20.5	13.0	1.00	106.0	760.0
4	2 40	10.18	221.4	221.1	177.2	192.6	28.5	28.7	141.8	142.5	142.1	340	371	711	17.4	40.0	40.1	20.8	15.0	.88	134.2	845.2
5	2 00	10.2	219.8	220.1	178.5	191.7	28.6	28.7	142.2	142.6	142.4	322	359	681	17.2	38.0	32.0	19.9	16.0	.95	124.0	805.0
1	4 00	10.33	218.1	220.2	166.6	192.4	28.9	29.0	144.33	144.09	144.21	375	404	779	......	......	......	......	......	......	......	......
6	2 40	15.25	224.4	229.3	193.0	209.4	28.5	28.8	210.9	217.2	214.0	1,174	1,472	2,646	18.2	40.5	29.0	44.6	14.0	1.05	147.3	2,793.3
7	3 00	20.16	252.2	252.4	220.3	207.1	28.6	28.9	284.28	284.27	284.27	3,456	3,622	7,078	23.0	44.5	51.0	56.8	53.4	1.15	229.8	7,307.8
9	2 40	22.55	248.4	247.4	221.2	211.1	28.5	29.0	319.7	319.9	319.8	4,962	5,205	10,167	22.4	42.0	160.0	97.0	191.4	1.13	513.9	10,680.9
8	3 00	24.37	251.1	251.4	241.4	241.1	28.5	28.8	351.5	351.6	351.5	7,376	7,391	14,767	25.8	45.0	197.0	113.4	277.0	1.20	659.4	15,426.4

[a] As determined from counter readings at beginning and end of test.

TABLE G.—*Steam consumption tests (second series).*

IV.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

U. S. S. Salem.
(Tables 52-65.)

		Horsepower.			Condensed exhaust steam (pounds).										
		Machinery necessary to propulsion.			Per hour.						Per knot.				
Number of test.	Speed per hour in knots.	S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines. $\frac{f}{c}$	Per H. P. of all machinery necessary to propulsion. $\frac{g}{e}$	Used by main turbines. $\frac{f}{b}$	Used for all purposes except lift pumps. $\frac{h}{b}$	Used for all purposes. $\frac{k}{b}$	Auxiliaries exhaust into—
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
1	10.01	719.6	131.66	851.26	30,182	38,636	44,783	45,144	42.0	45.4	3,015.2	4,473.8	4,509.9	Engine room auxiliary condenser.
4	10.06	723.2	122.06	845.26	....	....	40,523	40,850	....	....	....	4,028.1	4,060.6	Third stage of forward turbine.
2	12.27	1,477.1	161.38	1,638.48	44,674	54,566	59,610	60,091	30.2	33.3	3,641.0	4,858.2	4,897.4	Engine room auxiliary condenser.
3	12.38	1,474.2	157.69	1,631.89	....	....	56,630	57,087	....	....	....	4,574.3	4,611.2	Third stage of forward turbine.
5	12.49	1,487.7	145.98	1,633.68	46,091	55,369	60,246	60,732	30.9	33.9	3,690.0	4,823.5	4,862.4	Engine room auxiliary condenser.
7	14.83	2,463.0	164.11	2,627.11	....	....	79,188	79,827	....	....	....	5,339.7	5,382.8	Third stage of forward turbine.
6	14.85	2,506.0	168.51	2,674.51	64,980	77,087	82,283	82,947	25.9	28.8	4,375.7	5,541.0	5,585.6	Engine room auxiliary condenser.
8	17.42	4,720.0	217.03	4,937.03	93,270	107,895	112,603	113,511	19.8	21.9	5,354.1	6,464.0	6,516.1	Do.
10	17.43	4,614.0	191.97	4,805.97	92,325	105,804	110,582	111,474	20.0	22.0	5,297.0	6,344.3	6,395.5	Do.
9	17.44	4,624.0	215.27	4,839.27	....	....	107,859	108,729	....	....	....	6,184.6	6,234.4	Third stage of forward turbine.
14	20.05	7,153.0	256.07	7,409.07	131,407	148,385	154,314	155,559	18.4	20.0	6,553.9	7,696.4	7,758.5	Engine room auxiliary condenser.
13	22.47	11,154.0	352.35	11,506.35	182,887	204,691	209,592	211,282	16.4	17.8	8,139.1	9,327.6	9,402.8	Do.
12	23.98	13,954.0	479.29	14,433.29	236,789	266,097	270,259	272,439	17.0	18.4	9,874.4	11,270.0	11,361.0	Do.
11	25.12	18,070.0	681.92	18,751.92	292,751	332,373	338,754	341,486	16.2	17.7	11,654.1	13,485.4	13,594.0	Do.

TABLE G.—*Steam consumption tests (second series)*—Continued.

IV.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage). Engine room. Forward.	Steam pressure (gage). Engine room. After.	Steam pressure (gage). Steam chest. Forward.	Steam pressure (gage). Steam chest. After.	Vacuum (inches of mercury). Forward engine room.	Vacuum (inches of mercury). After engine room.	Revolutions (per minute).[a] Forward starboard.	Revolutions (per minute).[a] After port.	Revolutions (per minute).[a] Mean.	Horsepower. Machinery necessary to propulsion. S. H. P. main turbines. Forward starboard.	S. H. P. main turbines. After port.	S. H. P. main turbines. Both.	I. H. P. auxiliaries. Wet vacuum pumps.	I. H. P. auxiliaries. Dry vacuum pumps.	I. H. P. auxiliaries. Main circulating pumps.	I. H. P. auxiliaries. Main feed pumps.	I. H. P. auxiliaries. Forced-draft blowers.	I. H. P. auxiliaries. Oil pumps.	I. H. P. auxiliaries. Total (r+s+t+u+v+w).	Grand total (q+x).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y
	H. m.																					
1	2 00	10.01	262.0	261.0	154.7	121.1	28.2	28.5	138.7	140.0	139.3	354.9	364.7	719.6	15.8	34.4	45.6	24.7	10.1	1.06	131.66	851.26
4	2 00	10.06	256.8	254.6	207.0	188.0	28.6	28.6	140.0	140.0	140.0	358.3	364.9	723.2	17.0	33.0	36.0	23.0	12.3	.76	122.06	845.26
2	3 20	12.27	249.8	249.1	215.3	206.2	28.2	28.6	173.5	174.0	173.7	721.4	755.7	1,477.1	16.7	35.0	60.0	27.0	21.8	.88	161.38	1,638.48
3	2 00	12.38	251.9	250.4	217.4	206.6	28.2	28.6	173.5	173.3	173.4	721.4	752.8	1,474.2	16.8	34.4	61.4	27.5	16.8	.79	157.69	1,631.89
[b]5	2 00	12.49	200.3	201.0	30.7	30.0	28.7	28.6	174.9	175.0	175.0	727.5	760.2	1,487.7	17.4	29.4	51.4	25.8	21.2	.78	145.98	1,633.68
7	2 00	14.83	244.8	243.3	171.3	190.8	28.8	28.7	208.7	208.7	208.7	1,200.0	1,263.0	2,463.0	17.2	32.5	30.0	48.5	35.0	.91	164.11	2,627.11
6	2 40	14.85	250.3	248.8	173.4	200.6	28.9	28.6	208.9	208.9	208.9	1,236.0	1,270.0	2,506.0	17.5	32.5	34.4	49.6	33.6	.91	168.51	2,674.51
8	2 40	17.42	245.5	244.0	192.0	207.7	28.7	28.4	246.4	246.5	246.4	2,365.0	2,355.0	4,720.0	21.2	33.5	52.6	63.6	45.2	.93	217.03	4,937.03
10	2 40	17.43	250.0	249.5	192.0	204.0	28.8	28.7	246.4	246.5	246.5	2,276.0	2,338.0	4,614.0	19.3	30.0	42.0	57.6	42.0	1.07	191.97	4,805.97
9	2 00	17.44	250.4	248.4	178.0	197.7	28.7	28.7	246.7	246.7	246.7	2,314.0	2,310.0	4,624.0	23.8	33.5	54.0	60.6	42.4	.97	215.27	4,839.27
14	2 20	20.05	240.0	238.0	187.7	199.4	28.7	28.6	285.1	285.2	285.2	3,485.0	3,668.0	7,153.0	22.4	32.9	71.0	68.5	60.0	1.27	256.07	7,409.07
13	2 00	22.47	246.7	242.4	201.4	200.3	28.6	28.4	322.2	322.2	322.2	5,463.0	5,691.0	11,154.0	23.5	32.0	100.5	85.0	110.0	1.35	352.35	11,506.35
12	2 00	23.98	259.0	253.0	198.0	200.4	28.3	28.1	347.3	347.2	347.2	6,921.0	7,033.0	13,954.0	25.7	33.8	104.0	115.5	198.9	1.39	479.29	14,433.29
11	2 00	25.12	264.7	259.4	249.8	255.7	28.2	28.1	369.9	369.7	369.8	9,080.0	8,990.0	18,070.0	28.7	34.0	135.6	139.2	343.0	1.42	681.92	18,751.92

[a] As determined from counter readings at beginning and end of test.

[b] All nozzles open, speed controlled by throttle.

Particular attention is invited to results of tests Nos. 2 and 5 of the second series, which were run at about the same speed of the vessel. Test No. 5 was made with all (20) nozzle valves open, regulation being by turbine throttles entirely, for the sole purpose of comparing steam consumption under these conditions with that in test No. 2, in which but 3 (after) and $3\frac{3}{16}$ (forward) nozzles were open. In the latter test (which is the ordinary method of operation) turbine throttles were practically wide open, slight modifications in revolutions when necessary being made by manipulation of the throttle valves. It is to be noted further that there was wide difference in steam-chest pressures in the two tests, the average (per gage—both turbines) for No. 2 test being about 210 pounds and for No. 5 test about 30 pounds, while pressures in main steam line were approximately 249 and 201, respectively. As will be seen from an examination of the data, there was no appreciable difference in economy of the turbines in the two tests.

Suitable connections are installed to the third and fourth stages of main turbines for utilization of auxiliary exhaust steam. In test No. 4 (first series) and tests Nos. 3, 4, 7, and 9 (second series) the auxiliary exhaust line was open to feed heaters as well as to third stages of turbines.

CHESTER.

[Tables 66 to 94.]

Results of economy tests on turbines of the Parsons type are influenced to a great extent by the amount of clearance allowed between dummy pistons and rings which controls steam leakage. On the *Chester* adjustments of dummy clearances are made by movement of the lower and upper halves of thrust blocks, and the operation is one requiring both skill and accuracy. Upper half of thrust blocks are intended to take the steam thrust only, while the lower half absorbs the propeller thrust. For the two cruising turbines, which are on inboard shafts (Nos. 2 and 3) and located forward of each main L.P. turbine, expansion couplings are installed, the arrangement being such that thrust bearings of these turbines are not subjected to propeller thrust. All main turbines are designed to be balanced, the end thrust on the rotor and the unbalanced thrust on the blading, both resulting from steam pressure, being equal to and in the opposite direction to the propeller thrust. Perfect equilibrium, however, under all conditions of operation is practically impossible, and while for some speeds and turbine combinations propeller thrust predominates, others occur in which the steam pressure overbalances the propeller thrust. It will be apparent that in either case there is a tendency to a fore-and-aft movement (technically known as "float") of shaft and its attached rotor, which is forward in event of propeller thrust preponderating and aft under reverse conditions, or when steam thrust overbalances propeller thrust. The amount of "float," although small in any case, bears directly upon dummy clearances, operating to enlarge or reduce the same, depending upon whether the shaft movement is forward or aft. Moreover, these clearances, which are determined by micrometer gages, fitted permanently to the casing of each turbine, vary with temperature changes. Reliable readings can only be secured, therefore, under normal working conditions.

No effort was made while steam-consumption trials were in progress to adjust dummy clearances closer than usual service practice, since it was the aim to obtain steam economy under ordinary cruising conditions and not ideal results. A summary of such clearances, as measured while the various steam-economy tests were being run, is as follows:

SIX TURBINES.

Speed.	H. P. C.	I. P. C.	S. H. P.	S. L. P.	P. H. P.	P. L. P.
10 knots	0.021	0.026	0.021	0.015	0.022	0.030
12½ knots	.023	.025	.016	.012	.022	.025
15 knots	.023	.028	.015	.014	.027	.027
16 knots	.023	.029	.019	.015	.027	.024
18 knots	.031	.033	.018	.008	.027	.020

FIVE TURBINES.

Speed.	H. P. C.	I. P. C.	S. H. P.	S. L. P.	P. H. P.	P. L. P.
10 knots [a]	....	....	....	....	....	....
15 knots [a]	....	....	....	....	....	....
20 knots	....	0.019	0.007	0.006	0.023	0.016
22.50 knots	....	.023	.016	.011	.026	.016

[a] No readings taken.

FOUR TURBINES.

Speed.	H. P. C.	I. P. C.	S. H. P.	S. L. P.	P. H. P.	P. L. P.
10 knots	....	....	.018	.041	.017	.051
15 knots	....	....	.022	.012	.024	.026
20 knots	[a] 0.017	[a] .018	.020	.016	.034	.027
22.50 knots	[a] .013	[a] .020	.017	.018	.033	.025
24 knots	....	....	.026	.019	.036	.024
24.75 knots	....	....	.024	.018	.031	.028

[a] Cruising turbines revolving idly.

Brief mention has been made in describing the *Chester's* turbines of the means employed to prevent leakage (in or out) around shafts where the latter pass through turbine-casing heads. Plate 7 shows sectional views of the H. P. and L. P. turbine gland boxes, from which, it will be observed, that the packing consists, essentially, of a number of rows of knife-edge strips, supplemented by brass packing rings. The function of gland boxes is to check leakage of steam into the engine rooms when the pressure is greater in the rotor casings than the atmosphere, or air leakage into the turbines in event of the pressure being below the atmosphere, which latter, unless prevented, would cause a noticeable vacuum drop. Conditions arise, with the various turbine combinations, such that certain gland boxes (8 of the 12 provided) are required to pack against a steam pressure out, while under other conditions of operation these same gland boxes must be practically tight against an air leakage in. In operating the 6-turbine combination, to cite an example, gland boxes of cruising turbines are required to pack against a steam pressure in rotor casings, which usually is above the atmosphere. When, however, the 4-turbine combination is in use, both cruising turbines revolve idly

in a vacuum (drains to these turbines being open to the main condenser) and their gland boxes are required to be tight against a leakage of air into the turbine casings. Generally speaking, gland boxes of initial turbines are required to resist steam leakage out, but in all cases the function of the gland boxes of L. P. turbines is invariably to prevent air leakage into the casings.

Glands of all turbines are connected by branches to a system of piping, with suitable cut-out valves extending the length of both engine rooms. When steaming, it is the aim to maintain in this system at all times a pressure of about 1 pound per gage, and this may be effected through a branch connection to the auxiliary exhaust line, or in case steam leakage out from any gland box is considerable in amount use of such steam is made for this purpose. With the type of gland box installed on the *Chester* steam leakage depends largely upon the highest final pressure existing in any turbine casing, and at full power, with the 6-turbine combination in use, this pressure (about 102 pounds absolute—see test No. 8) becomes a maximum, being approximately the same as the initial pressure in the I. P. cruising turbine. Under such circumstances (and as well other powers of the 5 and 6 turbine combinations) the gland leakage is more than sufficient to pack the entire gland system, and unless provision were made to dispose of such excess steam an intolerable condition would soon result in the forward engine room. A 3-inch branch pipe, with cut-out valve, connects the gland system with the main condenser, and through this any surplus leakage is delivered. It will be noted that steam thus condensed is not fully utilized in the turbines, and from the standpoint of economy it would seem that a branch connection from the gland system to one or more stages of the low-pressure turbines would be a desirable addition.

Synopsis of results of steam-consumption trials, arranged in accordance with turbine combinations (6, 5, and 4), are shown on Tables H, K, and L.

TABLE H.—*Steam consumption tests (6 turbines).*

IV.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

U. S. S. Chester.
(Tables 66 to 94.)

		Horsepower.			Condensed exhaust steam (pounds).									
		Machinery necessary to propulsion.			Per hour.						Per knot.			
Number of test.	Speed per hour in knots.	S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines. $\frac{f}{c}$	Per H. P. of all machinery necessary to propulsion. $\frac{g}{e}$	Used by main turbines. $\frac{f}{b}$	Used for all purposes except lift pumps. $\frac{h}{b}$	Used for all purposes. $\frac{k}{b}$	Auxiliaries exhaust into—
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
a1	10.03	868.4	118.2	986.6	....	....	....	....	....	....	....	....	....	Engine room auxiliary condenser.
27	10.15	908.0	152.0	1,060.0	25,492	37,641	43,411	43,761	28.1	35.5	2,511.5	4,276.9	4,311.4	Do.
a2	10.65	945.6	121.5	1,067.1	....	....	....	....	....	....	....	....	....	Second stage main H. P. turbines.
a3	12.25	1,758.6	172.4	1,931.0	....	....	....	....	....	....	....	....	....	Engine room auxiliary condenser.
a4	12.75	1,881.7	157.5	2,039.2	....	....	....	....	....	....	....	....	....	Second stage main H. P. turbines.
24	13.00	1,944.9	178.9	2,123.8	41,208	54,993	60,842	61,333	21.2	25.9	3,169.8	4,680.2	4,717.9	Engine room auxiliary condenser.
22	14.80	3,118.4	191.2	3,309.6	55,728	70,249	75,975	76,588	17.9	21.2	3,765.4	5,133.4	5,174.9	Do.
6	15.00	3,131.8	144.2	3,276.0	....	....	74,170	74,768	....	....	....	4,944.7	4,984.5	Second stage main H. P. turbines.
a5	15.15	3,131.2	168.6	3,299.8	....	....	....	....	....	....	....	....	....	Engine room auxiliary condenser.
23	15.92	3,919.9	200.2	4,120.1	64,605	79,768	85,204	85,891	16.5	19.4	4,058.1	5,352.0	5,395.2	Do.
26	17.67	5,455.2	253.2	5,708.4	84,988	102,392	111,577	112,477	15.6	17.9	4,809.8	6,314.5	6,365.4	Do.
7	18.03	5,699.2	199.0	5,898.2	94,082	109,279	114,534	115,458	16.5	18.5	5,218.1	6,352.4	6,403.7	Do.
8	18.15	5,977.5	184.8	6,162.3	....	....	112,645	113,490	....	....	....	6,206.3	6,252.9	L. P. turbines.

[a] Steam consumption of this test is not considered reliable.

TABLE H.—*Steam consumption tests (6 turbines)*—Continued.

IV.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage). Forward engine room.	Steam pressure (gage). H. P. cruising steam chest.	Vacuum (inches of mercury). Forward engine room.	Vacuum (inches of mercury). After engine room.	Revolutions (per minute).[a] Starboard mean shafts Nos. 1 and 2.	Revolutions (per minute).[a] Port mean shafts Nos. 3 and 4.	Revolutions (per minute).[a] Mean of all shafts.	Horsepower. Machinery necessary to propulsion. S. H. P. main turbines. Starboard shafts 1+2.	Port shafts 3+4.	All shafts.	I. H. P. auxiliaries. Augmentors.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced-draft blowers.	Oil pumps.	Total (p+q+r+s+t+u).	Grand total (o+v).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w
	H. m.																			
1	5 00	10.03	230.8	50.0	28.8	28.8	210.9	205.8	208.3	444.0	424.4	868.4	46.5	14.7	22.2	27.0	7.2	0.62	118.2	986.6
27	2 00	10.15	223.0	50.0	29.0	29.6	207.6	205.1	206.3	437.6	470.4	908.0	46.4	27.5	42.8	31.8	2.5	.96	152.0	1,060.0
2	3 00	10.65	232.7	50.0	28.8	28.3	221.9	212.2	217.1	495.9	449.7	945.6	47.3	17.1	23.8	26.1	6.6	.57	121.5	1,067.1
3	2 40	12.25	229.2	105.2	28.5	29.3	256.8	250.6	253.6	914.0	844.6	1,758.6	46.8	21.1	43.2	48.2	12.4	.72	172.4	1,931.0
4	2 40	12.75	231.9	104.9	28.5	29.3	267.1	260.0	263.6	988.0	893.7	1,881.7	47.0	20.8	37.3	34.3	17.4	.74	157.5	2,039.2
24	2 00	13.00	230.5	105.0	29.0	29.4	271.2	265.6	268.4	1,034.3	910.6	1,944.9	47.8	28.8	54.0	38.5	9.0	.78	178.9	2,123.8
22	2 00	14.80	223.5	150.0	28.8	29.4	310.9	304.8	307.8	1,592.8	1,525.6	3,118.4	45.9	29.0	64.0	40.0	11.5	.78	191.2	3,309.6
6	2 40	15.00	229.3	151.0	28.3	28.9	315.2	309.8	312.5	1,623.9	1,507.9	3,131.8	43.3	15.1	25.6	41.5	18.0	.68	144.2	3,276.0
5	2 40	15.15	232.0	151.0	28.3	29.0	318.6	313.6	316.1	1,611.2	1,520.0	3,131.2	47.4	18.6	31.2	53.3	17.4	.74	168.6	3,299.8
23	2 00	15.92	226.8	180.0	28.8	29.4	335.7	329.5	332.6	2,031.2	1,888.7	3,919.9	47.2	30.3	66.8	41.5	13.6	.81	200.2	4,120.1
26	2 00	17.67	243.0	235.0	29.0	29.6	372.9	369.6	371.2	2,761.5	2,693.7	5,455.2	50.8	31.2	101.0	51.0	18.0	1.17	253.2	5,708.4
7	2 40	18.03	243.9	235.0	28.5	29.4	380.5	377.5	379.0	2,909.5	2,789.7	5,699.2	50.6	16.3	40.0	72.6	18.0	1.49	199.0	5,898.2
8	2 00	18.15	247.5	235.0	28.5	29.4	383.1	379.5	381.3	3,016.0	2,961.5	5,977.5	51.4	14.3	32.0	64.5	21.0	1.59	184.8	6,162.3

[a] As determined from counter readings at beginning and end of test.

TABLE K.—*Steam consumption tests (5 turbines).*

IV.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower. Machinery necessary to propulsion.			Condensed exhaust steam (pounds). Per hour.						Condensed exhaust steam (pounds). Per knot.			Auxiliaries exhaust into—
		S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines. $\frac{f}{c}$	Per H. P. of all machinery necessary to propulsion. $\frac{g}{e}$	Used by main turbines. $\frac{f}{b}$	Used for all purposes except lift pumps. $\frac{h}{b}$	Used for all purposes. $\frac{k}{b}$	
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
9	10.43	828.5	126.8	955.3	33,927	44,347	47,993	48,380	40.9	46.4	3,252.8	4,601.4	4,638.5	Engine room auxiliary condenser.
10	10.56	826.4	125.1	951.5	....	....	46,164	46,522	....	....	....	4,371.6	4,405.5	Second stage main high-pressure turbines.
12	15.10	3,053.6	154.2	3,207.8	63,565	75,818	80,815	81,467	20.8	23.6	4,209.6	5,352.0	5,395.2	Engine room auxiliary condenser.
13	15.35	3,169.5	146.9	3,316.4	....	....	77,660	78,286	....	....	....	5,059.3	5,100.0	Low-pressure turbines.
16	20.00	8,154.3	245.7	8,400.0	119,360	137,376	140,100	141,230	14.6	16.4	5,968.0	7,005.0	7,061.0	Engine room auxiliary condenser.
17	20.14	8,368.8	245.1	8,613.9	....	....	136,649	137,751	....	....	....	6,785.0	6,839.6	Low-pressure turbines.
20	22.62	12,467.8	309.8	12,777.6	159,435	181,336	186,430	187,933	12.8	14.2	7,048.4	8,241.8	8,308.3	Engine room auxiliary condenser.

TABLE K.—*Steam consumption tests (5 turbines)*—Continued.

IV.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

			Steam pressure (gage).		Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.											
										Machinery necessary to propulsion.											
										S. H. P. main turbines.			I. H. P. auxiliaries.								
Number of test.	Duration of test.	Speed per hour in knots.	Forward engine room.	I. H. P. cruising turbines.	Forward engine room.	After engine room.	Starboard mean shafts Nos. 1 and 2.	Port mean shafts Nos. 3 and 4.	Mean of all shafts.	Starboard shafts, 1+2.	Port shafts, 3+4.	All shafts (m+n).	Augmentors.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced draft blowers.	Oil pumps.	Total (p+q+r+s+t+u).	Grand total (o+v).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w
	H. m.																			
9	2 40	10.43	230.1	15.0	29.2	29.5	206.7	217.7	212.2	415.0	413.5	828.5	46.8	18.1	20.1	32.3	9.0	.51	126.8	955.3
10	2 40	10.56	232.0	15.0	29.3	28.8	213.2	217.2	215.2	412.3	414.1	826.4	47.1	17.9	18.0	31.0	10.5	.61	125.1	951.5
12	3 00	15.10	229.3	60.0	29.3	29.3	306.4	322.5	314.5	1,319.4	1,734.2	3,053.6	46.7	22.2	38.0	36.5	10.0	.81	154.2	3,207.8
13	2 20	15.35	222.7	60.0	29.3	29.7	311.1	329.3	320.2	1,336.9	1,832.6	3,169.5	44.7	22.6	30.0	37.4	11.5	.74	146.9	3,316.4
16	2 40	20.00	228.0	145.0	29.0	29.6	408.3	435.5	421.9	3,378.1	4,776.2	8,154.3	46.9	29.8	87.0	71.0	10.0	.98	245.7	8,400.0
17	2 40	20.14	228.4	145.0	29.0	29.6	411.8	438.5	425.2	3,529.6	4,839.2	8,368.8	47.2	29.7	82.0	76.8	8.5	.86	245.1	8,613.9
20	2 40	22.62	230.4	210.0	28.5	29.4	465.9	496.9	481.4	5,404.7	7,063.1	12,467.8	48.5	31.1	110.0	92.0	27.0	1.16	309.8	12,777.6

[a] As determined from counter readings at beginning and end of test.

TABLE L.—*Steam consumption tests (4 turbines).*

IV.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

		Horsepower.			Condensed exhaust steam (pounds).									
		Machinery necessary to propulsion.			Per hour.						Per knot.			
Number of test.	Speed per hour in knots.	S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines. $\frac{f}{c}$	Per H. P. of all machinery necessary to propulsion. $\frac{g}{e}$	Used by main turbines. $\frac{f}{b}$	Used for all purposes except lift pumps. $\frac{h}{b}$	Used for all purposes. $\frac{k}{b}$	Auxiliaries exhaust into—
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
11	10.15	746.6	143.6	890.2	33,539	45,544	47,291	47,690	44.9	51.2	3,304.3	4,659.2	4,698.5	Engine room auxiliary condenser.
15	15.68	3,463.8	180.3	3,644.1	....	....	93,864	94,621	....	....	....	5,986.2	6,034.5	L. P. turbines.
14	15.81	3,613.0	183.6	3,796.6	77,252	93,172	94,880	95,621	21.4	24.5	4,886.3	6,001.2	6,048.1	Engine room auxiliary condenser.
18	20.05	7,834.7	284.8	8,119.5	132,651	151,585	156,954	158,220	16.9	18.7	6,616.0	7,828.0	7,891.3	Do.
19	20.05	7,849.5	297.3	8,146.8	....	....	154,749	155,997	....	....	....	7,718.0	7,780.9	L. P. turbines.
21	22.78	12,156.4	338.8	12,495.2	176,551	198,803	203,909	205,553	14.5	15.9	7,750.3	8,951.2	9,023.4	Engine room auxiliary condenser.
29	23.85	15,651.0	545.9	16,196.9	215,117	247,473	252,164	254,198	13.7	15.3	9,019.6	10,572.9	10,658.2	Do.
25	23.92	15,185.8	385.9	15,571.7	221,173	246,907	254,783	256,838	14.6	15.9	9,246.3	10,651.5	10,737.4	Do.
28	24.67	18,544.0	606.9	19,150.9	253,662	290,861	293,687	296,054	13.7	15.2	10,282.2	11,904.6	12,000.6	Do.

TABLE L.—*Steam consumption tests (4 turbines)*—Continued.

IV.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

			Steam pressure (gage).				Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.										
												Machinery necessary to propulsion.										
			Engine room.		Steam chest.							S. H. P. main turbines.			I. H. P. auxiliaries.							
Number of test.	Duration of test.	Speed per hour in knots.	Forward.	After.	Forward main H. P.	After main H. P.	Forward engine room.	After engine room.	Starboard mean shafts Nos. 1 and 2.	Port mean shafts Nos. 3 and 4.	Mean of all shafts.	Starboard shafts 1+2.	Port shafts 3+4.	All shafts (o+p).	Augmenters.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced draft blowers.	Oil pumps.	Total (r+s+t+u+v+w).	Grand total (x+y).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y
	H. m.																					
11	2 40	10.15	226.8	231.5	6.0	6	29.1	29.4	205.9	207.8	206.8	402.4	344.2	746.6	46.3	18.3	23.0	46.5	9.0	0.55	143.6	890.2
15	2 00	15.68	225.8	231.2	42.0	41	28.6	29.6	325.1	329.0	327.1	1,716.8	1,747.0	3,463.8	46.6	25.2	22.5	60.2	25.0	.78	180.3	3,644.1
14	2 40	15.81	229.0	230.6	42.0	41	29.0	29.6	329.0	331.8	330.4	1,776.0	1,837.0	3,613.0	46.7	24.7	24.0	64.0	23.5	.74	183.6	3,796.6
18	2 40	20.05	217.7	220.7	87.5	87	28.8	29.6	422.3	421.8	422.1	3,939.7	3,895.0	7,834.7	44.6	26.7	123.0	79.0	10.5	.98	284.8	8,119.5
19	2 20	20.05	219.6	224.7	87.5	87	28.8	29.5	422.6	421.9	422.3	3,931.9	3,917.6	7,849.5	45.5	28.8	130.0	82.0	10.0	1.04	297.3	8,146.8
21	2 40	22.78	222.2	228.9	126.0	126	28.5	29.2	484.4	481.2	482.8	6,088.5	6,067.9	12,156.4	46.1	27.7	136.0	90.0	37.5	1.52	338.8	12,495.2
29	1 00	23.85	220.0	232.75	156.0	156	28.0	29.0	513.9	509.1	511.5	7,849.7	7,801.3	15,651.0	47.1	45.2	246.0	118.8	87.0	1.79	545.9	16,196.9
25	1 00	23.92	224.5	230.5	156.0	156	28.1	28.9	515.5	512.6	514.1	7,768.8	7,417.0	15,185.8	47.3	28.9	118.0	111.6	78.9	1.22	385.9	15,571.7
28	2 00	24.67	228.3	241.3	185.0	185	28.0	28.8	544.8	540.2	542.5	9,365.0	9,179.0	18,544.0	48.1	57.7	276.0	121.5	101.5	2.14	606.9	19,150.9

[a] As determined from counter readings at beginning and end of test.

PERFORMANCE CURVES.

From data of steam consumption tests, a number of curves showing performances have been laid down, as described below:

(*a*) Speed in knots—pounds of water per hour:
Plates: *Birmingham*, 57; *Salem*, 58 (first series) and 59 (second series); *Chester*, 60.

(*b*) Speed in knots—pounds of water per knot:
Plates: *Birmingham*, 61; *Salem*, 62 (second series); *Chester*, 63.

(*c*) Horsepower—pounds of water per horsepower:
Plates: *Birmingham* 64; *Salem*, 65 (first series) and 66 (second series); *Chester*, 67.

(*d*) Pounds of water per shaft horsepower—shaft horsepower, 1 turbine:
This shows comparison of steam consumption of *Salem's* turbines before and after overhauling; each turbine plotted separately. Plate 68.

(*e*) Pounds of water per hour per revolution—revolutions per minute:
This shows comparison of steam consumption of *Salem's* turbines before and after overhauling; each turbine plotted separately. Plate 69.

(*f*) Speed in knots—pounds of water (total) per hour:
All vessels. Plate 70.

(*g*) Speed in knots—pounds of water (main engines or turbines) per hour:
All vessels. Plate 71.

(*h*) Speed in knots—pounds of water (total) per knot:
All vessels. Plate 72.

(*k*) Speed in knots—pounds of water (main engines or turbines) per knot:
All vessels. Plate 73.

V. COAL CONSUMPTION TESTS AT SPEEDS APPROXIMATELY OF 10, 15, AND 20 KNOTS.

VI. COAL CONSUMPTION TEST AT FULL POWER.

[Tables for Birmingham, 95–98; Salem, 99–102; Chester, 103–106.]

Four comparative coal-consumption tests were carried out as outlined above, the vessels being in company, and consequently subjected to the same weather conditions. As planned, these tests were to be of 100, 50, 100, and 24 hours' duration, respectively, but on account of rough weather the 10-knot trial was discontinued after steaming 96 hours, and for a similar reason the 20-knot trial ended after 98 hours.

Bradford, R. I., was selected as an accessible and suitable base from which to operate, particularly as the same grade of coal (New River) was available at this coaling station for all trials. The use of the same grade of coal, it may be pointed out, is a desirable, if not a necessary, condition to accurate comparison.

Before each coaling an estimate was made for each vessel of the amount necessary to be taken on board, so that the mean displacement for each trial, as nearly as could be figured beforehand, would average 4,000 tons. This calculation included, in addition, an allowance for "make-up" feed water, which in all coal-consumption trials was carried in double-bottom compartments, the distilling plant being operated only to supply fresh water for purposes other than the boilers. The displacement, as finally determined, was figured from drafts taken at the beginning and end of each trial, corrected for weights removed while reaching the course, and in steaming to an anchorage after completion of the trial.

On coal-consumption tests the aim was to effect a comparison of the sea performance of the three vessels as measured by the coal consumed. As outlined originally, shifting of an engineers' personnel

(firerooms only) from ship to ship was contemplated, with a view of thus fixing with precision the number in the complement, and, in addition, eliminating as far as possible the personal factor in firing. This plan, however, was abandoned because of practical difficulties in successfully carrying out the scheme, and for the further reason that, under such circumstances, the various trials of all vessels would have had to be separately carried out, instead of steaming in company, with resulting complications as to influence of weather conditions. All trials were run, therefore, with the engineers' force regularly assigned to the vessels, the complements of which, with reference to numbers in each rating, differed somewhat; but it is impossible under the circumstances, to frame conclusions with accuracy covering the comparative efficiencies of the three engineering personnels.

No attempt was made to directly measure the steam used by either the main or auxiliary machinery, the water-measuring apparatus having been removed from all ships prior to commencement of these trials. Attention was concentrated primarily on securing an accurate determination of the quantity of coal consumed, although on all trials such detailed data were recorded as would enable a close calculation to be made of the steam used by any or all of the machinery in operation by comparison with previous steam-consumption tests.

As accuracy of comparison on trials of this character depends almost exclusively on the accuracy with which the quantity of coal burned is ascertained, the method of securing such data received careful attention when the trials were planned. It is obvious that any practical method to obtain such information adaptable to conditions on shipboard is apt to involve an element of doubt, and, therefore, the method selected should be the one least likely to error. The following were suggested and considered: First, to weigh all coal as taken from the bunkers; second, to use only coal put up in bags of known weight; third, to tally buckets or baskets of coal as taken from the bunkers, and at intervals to ascertain average weight; fourth, to mark the coal bunkers horizontally at intervals of 100 cubic feet with painted stripes, so that by leveling at the beginning and end of each trial the number of cubic feet of coal taken from each bunker could be calculated. Any of the four methods mentioned involve uncertainties as to correct weight, the most important reasons therefor being as follows: (1) The weighing of coal as taken from bunkers on account of the crowded condition of firerooms would hardly be a feasible scheme on the scout cruisers at high powers, no matter how many extra men might be assigned for the purpose; (2) it would have been expensive to bag coal in sufficient quantities for the proposed trials, and, furthermore, bagging material ordinarily used generally becomes injured in handling on shipboard, resulting in coal being spilled and for which it is difficult to make accurate allowance; (3) experience shows that calculations for coal expenditures by tallying buckets, even when great care is exercised, almost invariably fall short of the true weight, undoubtedly attributable to the tendency on part of the fireroom staff to "heap" buckets, in addition to the constant liability of error (generally one way only) in tallying, which latter, it may be noted, is a necessary adjunct to the first two methods; (4) in stowing of coal bunkers, there is never absolute certainty that all pockets or spaces have been properly filled, especially around beams, stiffeners, platforms, etc., and, moreover, the weight of coal which can

be stored in a given bunker is governed to some extent by the relative proportions of slack and lump.

The latter method was selected as being open to the fewest objections and as offering the best practical means of comparing results of coal-consumption trials. In this connection it may be stated that on preliminary government-acceptance trials of these vessels (for 24-hour trials at 12 and 22½ knots, respectively) it was the intention to make use of this method of determining coal consumption and the bunkers consequently were striped originally at the building yards. On the trials referred to, however, the *Chester* was the only ship on which the method was used.

At the Board's request the original bunker striping was subjected to critical examination, alterations and additions being made thereto under jurisdiction of the Bureau of Construction and Repair by the naval constructor attached to the Boston Navy-Yard, while the vessels were being fitted with water-measuring apparatus.

On the *Chester* the striping extended entirely around each bunker at intervals of 100 cubic feet, while on the *Birmingham* and *Salem* it was planned to carry only every fifth stripe around the bunkers, with intermediate divisions representing 100 cubic feet marked on ships' frames, bulkheads, etc., at frequent intervals, both in an athwartship and fore-and-aft direction. Curves were supplied the board by the Bureau of Construction and Repair showing capacities of the different bunkers, in accordance with the striping, which curves were used in all calculations of coal burned on the various trials.

The coal space on the *Birmingham* and *Salem* is divided into 21 separate coal bunkers, of which 1 (athwartship) bunker is forward, 12 abreast, and 8 abaft of the boiler compartments, each group being designated, respectively, by the letters A, B, and C, according to location. The coal-bunker arrangement of the *Chester* is similar to the other two ships, with the exception that there are but 6 instead of 8 (C) bunkers abaft the boiler compartments, and therefore a total of 19. All bunkers are connected by doors below the berth-deck level and a coal trolley is installed (portable through bulkhead-door openings) which when rigged furnishes the means of transporting coal from one bunker to another. The total bunker space in cubic feet available for coal stowage on each of the three vessels is as follows:

Bunkers.	Birmingham.	Salem.	Chester.
A compartments	4,906	4,906	4,859
B compartments	35,071	35,019	35,647
C compartments	20,072	19,771	19,974
Total	60,049	59,696	60,480

On all ships the space available for coal stowage in each of the 12 B bunker compartments varies from about 2,450 cubic feet as a minimum to about 3,600 cubic feet as a maximum, similarly located bunkers being approximately of the same capacity. In all trials coal was used from B bunkers only, with the exception that on the 20-knot trial it was necessary on the *Salem* to make use also of the coal stowed in the A bunker and the two forward C bunkers. The number of bunkers from which coal was taken during the various trials is shown in the following table:

Trial.	Birmingham.	Salem.	Chester.
10-knot	2	4	4
15-knot	4	4	4
20-knot	12	15	12
Full power	6	12	7

On all trials, comparison is based upon number of cubic feet of coal consumed. In order, however, to convert such results into units of weight, the cubical space contained in cars used in transporting coal from storehouses to the wharf when coaling was calculated and corresponding weights of coal ascertained. The cars used were small and approximately of 2 and 4 tons capacity. They had a plane side surface, the base being constructed with a triangular jog in the central part and extending the full length to facilitate emptying. As coaling proceeded, preliminary to each test, a number of cars in use were measured; each was weighed when empty, and again when loaded and leveled. The average of all calculations showed a ton of coal to occupy a space of 43.5 cubic feet, and although this figure seems somewhat high, it has been adopted in computing coal consumed in tons on all trials.

After coaling and prior to commencement of each trial, such bunkers (after leveling) as were to be used, were carefully inspected, and the distance noted from top of coal to the nearest stripe, from which data cubical contents were readily calculated. While the distance between adjacent stripes varied in the same bunker, and, of course, for different bunkers, the average was approximately 7 inches for those bunkers abreast the firerooms. When leveled, readings by independent observers rarely differed for any bunker by as much as 15 cubic feet, and this figure may be regarded as representing the maximum liability to personal error, assuming the stripes to have been properly located and the coal closely stowed. It should be pointed out here that the tendency of bunker coal is to settle when ships roll at sea, which tendency is pronounced at times and visibly noticeable, especially in bunkers indifferently stowed, where there has been failure to fill in properly around obstructions (girders, beams, platforms, etc.) when coaling. Directions were given, however, that particular care be exercised in this respect in stowing bunker coal for these trials, so that if instructions were carefully observed it will be evident that possible errors due to coal settling could not have been of moment in the results of any test, since all vessels were run under identical sea and weather conditions.

In general, the manner of securing coal data on the various trials was as follows: In steaming from the anchorage off Bradford to the open sea, and prior to commencement of each trial, coal was taken from bunkers other than those reserved for the trial. At the instant of starting, fireroom plates were swept clean, or an estimate made of the coal thereon, and doors raised on those bunkers from which coal was to be used. Upon completion of the trial, these bunker doors were lowered, care having been taken to leave no coal, or as little as possible, on the floor plates at that time. The coal remaining in the trial bunkers was then leveled and inspected, and the volume figured from the stripes, coal for the boilers in operation as the ship proceeded to port being taken from other bunkers. On the 20-knot

trial a slight modification was required, since it was necessary during this trial to make use of coal stowed in all B bunkers, and in consequence those bunkers from which coal was taken until the trial commenced were releveled and inspected.

During each trial a sample of coal was taken on the three vessels, and later subjected to analysis, as shown in Table M.

TABLE M.—*Coal consumption tests.*

COAL ANALYSIS.

Ship.	No. of test.	Proximate analysis. Coal as fired. Moisture.	Volatile matter.	Fixed carbon.	Ash.	Sulphur.	B. T. U.'s.	Dry coal. Volatile matter.	Fixed carbon.	Ash.	Sulphur.	B. T. U.'s.
Birmingham	1	1.39	20.91	70.80	6.90	0.86	14,311	21.20	71.80	7.00	0.87	14,513
	2	1.37	21.06	72.36	5.21	.90	14,559	21.35	73.37	5.28	.91	14,761
	3	1.27	20.79	72.22	5.72	.75	14,478	21.06	73.15	5.79	.76	14,664
	4	1.49	21.29	71.08	6.14	.69	14,409	21.61	72.16	6.23	.70	14,627
Salem	1	1.37	21.43	72.17	5.03	.77	14,598	21.73	73.17	5.10	.78	14,801
	2	2.43	21.02	66.76	9.79	.66	13,604	21.54	68.42	10.04	.67	13,943
	3	1.49	20.24	72.45	5.82	.73	14,417	20.55	73.55	5.90	.74	14,635
	4	1.53	20.50	71.56	6.41	1.07	14,390	20.82	72.67	6.51	1.08	14,614
Chester	1	1.27	20.65	72.81	5.27	.81	14,481	20.92	73.74	5.34	.82	14,667
	2	3.24	19.30	69.86	7.60	.71	13,958	19.95	72.20	7.85	.73	14,425
	3	1.42	20.10	73.51	4.97	.85	14,548	20.39	74.57	5.04	.86	14,758
	4	1.04	20.92	73.34	4.70	.74	14,716	21.14	74.11	4.75	.75	14,871
Mean B. T. U.'s for all coal used							14,372					14,607

During trials at 10, 15, and 20 knots each vessel regulated revolutions of main propelling machinery for the respective speeds mentioned in accordance with standardization (pls. 85, 86, and 88) curves. Generally a column-like formation was preserved, although there was no attempt to maintain position with exactness. Revolutions were varied occasionally, but only as was necessary to keep the vessels within a reasonable distance of one another. Such changes were always slight, and usually made at long-time intervals, so that no ship was handicapped—as often happens when steaming regularly in squadron—by constant throttle valve changes to regulate speed. A number of turns were made during the 10 and 15 knot trials; on the 20-knot run but one turn was made. It should be mentioned that at each turn the leading vessel took up position astern in the column, and that at such times navigators of the three vessels established their relative positions by stadimeter observations, which was also done at the commencement and end of each trial. Part of Table N has been compiled from results of such observations, and it is to be noted that distances covered by each vessel on the various trials, as figured from revolutions, when taken in conjunction with observed stadimeter data, indicates a considerable influence on speed, apparently attributable to some, or all, of the following: Trim of vessels; steering; weather conditions. Courses steered, as well as weather conditions, are recorded under "Remarks" following data sheets of the *Salem* for all coal trials.

TABLE N.—*Coal consumption tests.*

			a	b	c	d	e	f	g	h	j	k
Test No.	Date.	Name of vessel.	Duration of test.	Average revolutions during test.	Knots per hour, revolution speed curves.	a×c.	d(C.)—d (S.), Chester gained on Salem.	Chester gained on Salem by observation.	d(C.)—d (B.), Chester gained on Birmingham.	Chester gained on Birmingham by observation.	e—f.	g—h.
			Hours.				*Knots.*	*Knots.*	*Knots.*	*Knots.*	*Knots.*	*Knots.*
1	9.30 a. m. March 21, 1909, to 9.30 a. m. March 25, 1909	Birmingham	96	73.02	9.86	946.56						
		Salem	96	138.23	9.90	950.40	+12.48	− 9.00	+16.32	− 2.50	+21.48	+18.82
		Chester	96	203.21	10.03	962.88						
2	9.45 a. m. March 29, 1909, to 11.45 a. m. March 31, 1909	Birmingham	50	111.91	15.00	750.00						
		Salem	50	209.32	14.91	745.50	+ 3.50	+ 1.50	− 1.00	+ 4.00	+ 2.00	− 5.00
		Chester	50	312.04	14.98	749.00						
3	1 p. m. April 3, 1909, to 3 p. m. April 7, 1909	Birmingham	98	149.84	19.83	1,943.34						
		Salem	98	282.27	20.25	1,984.50	−34.30	+ 3.97	+ 6.86	+ 3.59	−38.27	+ 3.27
		Chester	98	419.86	19.90	1,950.20						
4	10.45 a. m. April 12, 1909, to 7.30 p. m. April 12, 1909	Birmingham	8¾	187.24	23.83	208.51						
		Salem	8¾	360.20	24.81	217.08	+ 3.24	+ 3.40	+11.81		− 0.16	
		Chester	8¾	563.47	25.18	220.32						
4	10.45 a. m. April 12, 1909, to 10.45 p. m. April 12, 1909	Birmingham	12	189.50	24.00	288.00						
		Salem	12	357.50	24.68	296.16	+ 6.60	[a]+ 5.92	+14.76	[a]+10.85	+ 0.68	+ 3.91
		Chester	12	562.70	25.23	302.76						
4	10.45 a. m. April 12, 1909, to 10.45 a. m. April 13, 1909	Birmingham										
		Salem	24	350.25	24.32	583.68	+18.24	[b]+18.67			− 0.43	
		Chester	24	557.91	25.08	601.92						

[a] By observation on Salem and Birmingham.

[b] Final relative position having been determined on the assumption that the observation at the end of first eight and three-fourths hours was correct and the revolutions of the Salem having been modified to conform to this.

The observations of all ships at the end of eight and three-fourths hours of full speed test agreed, but *Birmingham's* position at this time is not given in navigator's report.

Speed per hour for eight and three-fourths hours run.

Birmingham	23.83
Salem	24.81
Chester	25.18

Speed of *Chester*, 25.18 knots, minus speed of *Salem*, 24.81 knots, equals 0.37 knot.

Chester gained on the *Salem* by observation 3.4 $\therefore \frac{3.4}{8.75}$ hours=0.3886=actual loss in distance by *Salem* per hour during first eight and three-fourths hours. On this basis then actual speed or distance traveled by *Salem* per hour=24.81+(0.37−0.3886=−0.0186)=24.7914 knots during first eight and three-fourths hours.

Nominal speed of *Salem* during twenty-four hours, determined by speed revolution curve=24.32

Applying correction found during eight and three-fourths speed of *Salem* during twenty-four hours=$24.32 \times \frac{24.7914}{24.81}$=24.302.

Speed per hour of *Chester*	25.08 knots.
Corrected speed per hour of *Salem*	24.302 knots.
	.778
	24 hours.
Distance *Chester* gained on *Salem* during twenty-four hours	18.67

On full-power run revolutions were not set, all vessels being permitted to proceed on a course previously laid out at the highest speed possible to maintain. A number of 180° turns were made, in which way all vessels were brought within reasonable distances of one another when too widely separated. For convenience in analyzing results as to speed, this trial has been divided into three periods in the table mentioned above. The first eight and three-fourths hours were selected as a period, because all ships were in agreement as to the distance the *Chester* had gained on the *Salem* by stadimeter observations. Proportionate values of the speeds obtained from the revolutions, based on the observations above, were used to establish the total observation distance gained by the *Chester* over the *Salem* during the whole twenty-four-hour run.

Due to a mishap to the machinery, the *Birmingham* was unable to continue this trial beyond twelve hours. The I. P. crosshead of forward propelling engine developed a slight knock in wrist-pin brasses after the trial had been in progress about four hours, which increased gradually, resulting (about 7 p. m.) in carrying away oiling gear to this crosshead. About 9.15 p. m., liners between brasses (inboard after-side) were thrown out, and thereafter the pounding noticeably increased, causing excessive vibration and detachment of oil cups on various parts of the engine. It was believed inadvisable and dangerous to continue the trial longer than the time stated, and accordingly permission was granted to stop the engines at 10.45 p. m. Upon examination it was found that several set screws to crosshead bolts of I. P. brasses had worked out and were missing; white metal of bearings protruded about $\frac{3}{8}$ inches beyond brasses, due to hammering action, and consequently wrist pins were very loose in bearings. A leak developed in receiver piping of the same engine—connection from H. P. to I. P. valve chests at the top—soon after trial began, but no serious inconvenience resulted therefrom, except abnormal loss of steam and consequent increase of "make-up" feed water required as the trial progressed. The after (port) engine ran smoothly throughout the trial, and without accident, except carrying away of indicator motion on I. P. cylinder. After finishing twelve hours the *Birmingham* proceeded to port (Tompkinsville, Staten Island) under the after-engines alone, averaging about 110 revolutions, and making a speed of about 11 knots with a helm angle of about $2\frac{1}{2}$°.

Continuous records of speed were taken on all vessels by Nicholson recording logs; copies of charts so obtained follow data sheets for each vessel. The jagged line which often appears in these charts is not due to actual variation in speed, but probably to rolling and pitching of the vessels, as well as frequent changes of helm in an effort to maintain a straight course. Noticeably large and quick drops are due to changes of course; speed, as recorded on these charts, can only be regarded as an approximation.

Synopsis of results of coal consumption trials are collated in Table O. As will be seen, an attempt has been made, from the weight of coal consumed, to figure boiler evaportaion, as shown on these trials, for comparison with results of (II) evaporative tests. The total steam used by all machinery, as recorded in column p of the table, has been carefully calculated, due allowance being made for such auxiliaries or apparatus as were intermittently used, and which required steam

for their operation. It has been assumed, moreover, in this calculation (column p) that all "make-up" feed water expended (which in amount was determined from calibrated double-bottom compartments) was converted into steam, which assumption is probably not entirely correct. It will be noted, from an inspection of columns q and r (data being brought to the same basis for purposes of comparison), that a higher equivalent evaporation than that shown on boiler tests is recorded for the *Birmingham* on the 10-knot trial, and also for the *Chester* on the run at full power. No plausible explanation can be given for this, although it may be pointed out that all figures in column q are dependent upon conversion of coal, as measured, into weight, in accordance with data obtained when coaling for these trials.

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TABLE O.—*U. S. Scouts Birmingham, Salem, and Chester coal consumption tests.*

V-VI.—SYNOPSIS OF HORSE POWER, STEAM AND COAL CONSUMPTION.

[Tables 95 to 106.]

					Horsepower.					Steam.						Coal for all purposes per—							
					Machinery necessary to propulsion.						Make up feed.[a]			Reduced to water evaporated from and at 212° per pound of coal.		Day.		Hour.					
					Main engines or turbines.		Auxiliaries.	Total.															
Number of test.	Duration of test.	Name of ship.	Displacement.	Speed per hour in knots.[b]	I. H. P.	S. H. P.=.94×I. H. P.	I. H. P.	I. H. P. (f+h).	H. P. (g+h).	For all purposes per hour from curves of steam consumption tests.	Tons per day.	Pounds per hour.	Actual total on test in pounds (m+o).	Actual on this test.	As established by boiler evaporative tests.	Cubic feet.	Tons (43.5 cubic feet=1 ton).	Pounds.	Pounds per hour per knot.	Pounds per square feet of grate surface.	Per I. H. P. of machinery necessary to propulsion.	Per H. P. of machinery necessary to propulsion.	Auxiliaries exhaust into—
a	b	c	d	e	f	g	h	k	l	m[c]	n	o	p	q	r	s	t	u	v	w	x	y	z
1	*Hrs* 96	Birmingham..	3,994.0	9.86	960	902.4	61.4	1,021.4	963.8	32,871	10.10	942.2	33,813	11.60	11.12	1,398	32.14	2,999	304	17.02	2.94	3.11	Second receiver, both engines.
		Salem.........	4,022.0	9.90	......	668.4	115.1	....	783.5	44,674	11.66	1,088.3	45,762	9.52	10.85	2,342	53.85	5,026	508	21.67	....	6.41	E.R. auxiliary condenser, 40 minutes; main condenser, 29 hours 40 minutes; remainder third stage, forward turbine.
		Chester.......	4,026.9	10.03	......	730.0	105.1	....	835.1	36,097	10.97	1,023.8	37,121	9.54	9.64	1,830	42.06	3,926	391	16.92	....	4.70	Port L. P. turbine.

2	50	Birmingham..	4,001.9	15.00	3,263	3,067.2	89.2	3,352.2	3,156.4	67,438	13.90	1,297.3	68,735	10.87	11.20	3,098	71.21	6,646	443	14.32	1.98	2.105	Second receiver, both engines.
		Salem..........	3,964.5	14.91	......	2,609.2	170.2	....	2,779.2	84,280	12.12	1,131.2	85,411	9.06	10.86	4,665	107.23	10,008	671	21.56	....	3.60	Third stage, forward main turbine.
		Chester.......	3,967.8	14.98	......	3,201.3	128.1	....	3,329.4	74,596	13.20	1,232.0	75,828	9.73	9.50	3,887	89.36	8,341	557	20.54	....	2.51	Port L. P. turbine.
3	98	Birmingham..	3,960.0	19.83	7,575	7,120.5	188.7	7,763.7	7,309.2	129,176	26.10	2,436.0	131,612	9.62	10.90	6,675	153.45	14,322	722	20.58	1.84	1.96	Second receiver, both engines.
		Salem..........	4,030.5	20.25	......	6,883.3	329.0	....	7,212.3	163,111	17.51	1,634.2	164,745	9.54	10.60	8,788	202.03	18,855	931	27.09	....	2.61	Third stage, forward main turbine.
		Chester.......	3,970.3	19.90	......	8,374.0	206.2	....	8,580.2	134,270	16.80	1,568.0	135,838	9.55	9.44	7,086	162.90	15,205	760	21.84	....	1.77	Starboard L. P. turbine.
4	12	Birmingham..	4,059.0	24.00	14,828	13,938.4	535.9	15,363.9	14,474.3	271,545	120.4	11,237.3	282,782	8.42	9.52	16,344	375.72	35,070	1,461	50.38	2.28	2.42	Main condensers.
	24	Salem..........	4,028.0	24.32	......	14,630.0	695.8	....	15,325.8	293,103	45.63	4,258.3	297,361	8.36	9.25	18,060	415.15	38,749	1,593	55.74	....	2.53	Main condenser, 30 minutes; remainder fourth stage, main turbines.
	24	Chester.......	3,961.6	25.08	......	19,632.0	463.6	....	20,095.6	314,161	27.00	2,520.0	316,681	8.70	8.03	18,679	429.40	40,078	1,598	57.42	....	1.99	Starboard L. P. turbine.

a It is assumed that all the make-up feed was evaporated.
b Speed of each ship was taken from its standardization speed-revolution curve.
c Correction was made for the difference in the running of the auxiliaries of these tests and their corresponding steam consumption tests.

TABLE O.—*U. S. Scouts Birmingham, Salem, and Chester coal consumption tests*—Continued.

V-VI.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

					Steam pressure (gage).				Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.												
					Engine room.		Steam chest main engines or initial turbine.							Machinery necessary to propulsion.												Grand total (s+bb).
														Main engines or turbines S. H. P.[b]			Auxiliaries I. H. P.									
Number of test.	Duration of test.	Name of ship.	Speed per hour in knots.	Number of turbines in use.	Forward.	After.	Forward.	After.	Forward.	After.	Forward starboard.	After port.	Mean.	Forward starboard.	After port.	Both.	Augmentor.	Wet vacuum pumps.	Dry vacuum pumps.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced-draft blowers.	Oil pumps.	Total (t+u+v+w+x+y+z+aa).	
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y	z	aa	bb	cc
	Hrs.																									
1	96	Birmingham..	9.86	...	210.0	209.0	50.0	48.7	25.0	27.2	73.02	73.03	73.02	451.2	451.2	902.4	.....	.....	.....	11.9	17.8	19.0	12.7	.....	61.4	963.8
		Salem..........	9.90	2	217.9	220.4	145.3	152.0	29.6	29.1	138.23	138.23	138.23	328.6	339.8	668.4	.....	19.2	37.5	.....	24.0	25.3	8.0	1.01	115.1	783.5
		Chester........	10.03	6	179.0	189.0	59.0	......	28.8	29.0	204.04	202.36	203.20	392.0	338.4	730.4	39.4	.....	.....	19.1	17.0	29.1	.....	.55	105.1	835.1
2	50	Birmingham..	15.00	...	214.0	214.0	98.0	103.0	25.6	26.8	111.88	111.95	111.91	1,495.5	1,571.7	3,067.2	.....	.....	.....	12.9	20.0	30.7	25.6	.....	89.2	3,156.4
		Salem..........	14.91	2	220.9	219.8	174.8	177.7	28.9	29.0	209.30	209.30	209.30	1,397.8	1,211.4	2,609.2	.....	19.5	38.2	.....	34.0	57.4	20.0	.92	170.0	2,779.2
		Chester........	14.98	6	197.0	202.0	159.0	......	28.3	28.9	313.60	310.47	284.43	1,621.[illegible]	1,580.0	3,201.3	40.6	.....	.....	14.0	32.4	40.5	.....	.61	128.1	3,329.4
3	98	Birmingham..	19.83	...	209.6	206.4	173.7	172.0	23.6	26.7	149.84	149.84	149.84	3,616.2	3,504.3	7,120.5	.....	.....	.....	16.0	31.0	62.0	79.7	.....	188.7	7,309.2
		Salem..........	20.25	2	220.6	220.6	189.1	181.0	28.3	28.6	282.27	282.27	282.27	3,423.6	3,459.7	6,883.3	.....	22.0	37.4	.....	92.8	72.2	103.5	1.08	329.0	7,212.3
		Chester........	19.90	5	172.0	176.0	147.0	......	28.0	28.9	407.02	432.68	419.85	3,470.0	4,904.0	8,374.0	37.8	.....	.....	26.1	60.5	75.0	6.0	.82	206.2	8,580.2
4	12	Birmingham..	24.00	...	250.0	238.0	229.0	226.0	25.0	26.3	189.90	189.10	189.50	7,380.0	6,558.4	13,938.4	.....	.....	.....	24.4	48.0	153.5	310.0	.....	535.9	14,474.3
	24	Salem..........	24.32	2	231.0	234.0	209.0	204.4	28.0	29.0	342.93	357.56	350.25	6,990.0	7,640.0	14,630.0	.....	33.2	37.2	.....	94.0	101.0	429.0	1.42	695.8	15,325.8
	24	Chester........	25.08	4	212.0	216.0	204.0	201.0	28.3	29.1	560.90	555.10	558.05	10,005.0	9,627.0	19,632.0	45.8	.....	.....	29.6	128.4	156.6	101.7	1.52	463.6	20,095.6

[a] As determined from counter readings at beginning and end of test.

[b] S. H. P. of Birmingham = .94 × I. H. P.

A member of the Board was assigned to each vessel for general supervision of coal consumption trials.

PERFORMANCE CURVES.

From data of coal-consumption trials, curves have been laid down as described below:

(*a*) Pounds of coal per hour—speed in knots:
Plates: *Birmingham*, 74; *Salem*, 75; *Chester*, 76.
(*b*) Pounds of coal per knot—speed in knots:
Plates: *Birmingham*, 77; *Salem*, 78; *Chester*, 79.
(*c*) Pounds of coal per H. P.—H. P. all machinery necessary to propulsion:
All vessels: Plate 80.
(*d*) Pounds of water evaporated per pound of coal—pounds of coal burned per square foot of grate surface:
All vessels: Plate 81.

For purposes of comparison, curves showing standardization data have been drawn for each vessel, and upon the same sheet points have been plotted for power, speed, and revolutions, determined on steam and coal consumption trials, as follows:

(*e*) Horsepower—revolutions per minute.
Speed in knots—revolutions per minute:
Plates: *Birmingham*, 82; *Salem*, 83; *Chester*, 84.

ACCURACY OF DATA.

In conducting comparative trials of vessels while underway, it is not practicable to secure all data with the same degree of precision. On trials described in this report, the most reliable data are the revolutions per minute of the main propelling machinery, and for those auxiliaries equipped with counters; errors made for a period do not alter the mean for the whole test, so long as counter readings are accurately recorded at the beginning and end; chance of error in securing these two readings, which would materially affect results, is not great.

With regard to condensed water collected on steam-consumption trials, either in the main or auxiliary tanks, as the time interval was always recorded, there is little likelihood of mistake in the number of full tanks, and any error which might arise would, therefore, be confined to inaccurate readings of height of water in feed tanks at the beginning and end of a test, or of the measuring tanks at the end; probable errors due to this, when distributed over the whole trial, would be of no import. In proportioning steam used by individual auxiliaries on these trials, however, from condensed exhaust collected as a whole, performance curves (auxiliary machinery tests) have been taken as a guide, and results should only be regarded as close approximations.

For purposes of comparison, it is necessary to convert revolutions into speed, and this can be done with accuracy from standardization curves; errors can only arise from differences due to condition of ship's bottom, wind and sea, or irregular steering. The greatest variation in speed, as determined from revolutions corrected by stadimeter observations, occurred on coal-consumption test No. 1. As will be seen by reference to Table N, measuring speeds by revolutions taken from standardization curves, the *Salem* ran 950.4 knots

at a speed of 9.9 knots per hour, while the *Chester* during the same time made 962.88 knots at an average speed of 10.03, thus showing a gain for the latter vessel of 12.48 knots; results of stadimeter observations, however, showed an actual loss of 9 knots. Assuming that the *Chester* actually made 962.88 knots, the *Salem* must have made 971.88 knots, or a gain of 2.27 per cent. By a similar calculation in test No. 3, the *Salem* shows a loss of 1.95 per cent instead of a gain. In this connection, it may be further pointed out, that on the two standardizations of the *Salem*, the greatest difference in revolutions for the same speed was about 3.5 or about 1 per cent, while in horsepower required, as shown by torsion meters, this difference amounts to a much larger percentage. On steam-consumption tests, the maximum speed error is not believed to exceed 2 per cent, since all such trials were carried out during favorable weather only.

Gage pressures, temperatures, and revolutions of auxiliaries not fitted with counters, when readings are taken at fixed periods, are liable to inaccuracies because conditions may not remain precisely the same during the period; such inaccuracies enter into the final mean average of the test. Moreover, gages on fast-running machinery often fluctuate violently unless throttled to an extent which reduces pressure readings; spring gages are not satisfactory for reading small pressures, and especially when connected up with long crooked piping so as to locate them in a convenient place. Due to the latter, readings of stage pressures on the *Salem's* (first series) steam and coal consumption trials are too unreliable for use in calculations. Previous to the second series of steam consumption tests, large gages were installed, connected directly to the various stages, and, in addition, high-grade thermometers for determining pressures from temperatures were fitted, the two showing close agreement.

Power (indicated) as recorded for the *Birmingham* is considered fully as accurate as that usually secured on high-speed engines. Shaft horsepower, however, as determined by torsion-meter readings on the *Salem* and *Chester* is not altogether reliable for reasons heretofore stated. Liability to error, it is believed, varies approximately as the power, and while at low speeds such errors, generally speaking, may amount to as much as 20 per cent, at high powers, 2 per cent is probably the maximum. For the *Salem*, at low speeds, the error in power on some trials is no doubt greater than that stated, due in part to the fact that any error in reading torsion meters is exaggerated in percentage of the whole because of short length of scale from which such readings are necessarily taken.

CONCLUSIONS.

The following conclusions are based on results of tests described in this report:

MAIN PROPELLING MACHINERY.

Tables P and Q have been compiled from curves plotted on plates 70 and 71, and show at a glance for the three vessels, respectively, the weight of steam required per hour for all purposes (P) as well as the amount used by the main engines or turbines only (Q) for speeds from 10 to 25 knots, at knot intervals, together with a percentage comparison based on the *Chester's* 4-turbine combination.

An examination of plate 70 (Table P) shows that, based on total steam used for all purposes (excluding lift pumps), the *Birmingham* is the most economical of the scout vessels up to (a) 20.6 knots (50 per cent of designed power of main engines). Above that speed the *Chester*, using the 5-turbine combination, becomes the most economical, and at (b) 21.6 knots the 4-turbine combination is more economical than the *Birmingham's* reciprocating engine installation. Up to (c) 22.25 knots, the *Birmingham* is more economical than the *Salem* (second series of tests), but above this speed the *Birmingham* becomes the least economical of the three vessels. Due probably to excessive gland leakage, the *Chester's* 6-turbine combination is less economical than the 5-turbine combination above (d) 17.4 knots; the 5-turbine combination, up to the limit of its speed, is invariably more economical than the 4-turbine combination. The *Chester's* 4-turbine combination is less economical than the *Salem's* installation up to (e) 19.45 knots, but more economical above that speed.

Based (pl. 71 and Table Q) on steam used at various speeds per hour by the main propelling machinery only (exclusive of all auxiliaries) the figures above stated change but slightly. Using reference letters as in the previous paragraph, the figures become (a) 20.6, (b) 21.5, (c) 22.45, (d) 18.0, (e) 18.9.

TABLE P.—*Total steam consumption for all purposes (exclusive of lift pumps) at speeds from 10 to 25 knots, at knot intervals.*

COMPILED FROM PLATE No. 70.

Speed.	Steam used per hour (pounds).					Steam consumption in percentage, referred to Chester's 4-turbine combination as 100.				
	Birmingham.	Salem (second series).	Chester. Turbines in use.			Birmingham.	Salem (second series).	Chester. Turbines in use.		
			Six.	Five.	Four.			Six.	Five.	Four.
10 knots	34,000	44,000	43,200	45,500	46,600	73	94	93	98	100
11 knots	39,000	51,000	48,600	51,700	53,600	73	95	91	96	100
12 knots	44,700	58,200	54,700	58,100	61,200	73	95	89	95	100
13 knots	51,000	66,200	61,000	65,000	69,200	74	96	88	94	100
14 knots	58,000	74,700	68,200	72,300	77,700	75	96	88	93	100
15 knots	66,000	84,000	76,500	80,500	87,000	76	97	88	93	100
16 knots	75,300	94,200	86,500	89,500	97,000	78	97	90	92	100
17 knots	86,200	105,400	98,500	100,300	108,200	80	97	91	93	100
18 knots	98,600	118,200		111,400	120,000	82	98.5		93	100
19 knots	114,400	132,500		124,500	133,500	86	99		93	100
20 knots	134,000	150,000		138,500	148,500	90	101		93	100
21 knots	160,000	170,500		155,000	166,500	96	102		93	100
22 knots	192,000	195,000		173,500	187,000	103	104		93	100
23 knots	232,800	227,400			214,700	108	106			100
24 knots	285,500	273,000			254,800	112	107			100
24.5 knots		300,000			284,000		106			100
25 knots		329,000								

TABLE Q.—*Steam consumption of main engines or turbines only at speeds from 10 to 25 knots, at knot intervals.*

COMPILED FROM PLATE No. 71.

Speed.	Steam used per hour (pounds).					Steam consumption in percentage, referred to Chester's 4-turbine combination as 100.				
	Birmingham.	Salem (second series).	Chester. Turbines in use.			Birmingham.	Salem (second series).	Chester. Turbines in use.		
			Six.	Five.	Four.			Six.	Five.	Four.
10 knots	21,600	29,900	24,250	31,400	32,300	67	93	75	97	100
11 knots	26,200	36,100	30,200	36,900	38,900	67	93	78	95	100
12 knots	31,300	42,800	36,200	42,500	45,700	68	94	79	93	100
13 knots	36,900	50,000	42,500	48,800	53,000	70	94	80	92	100
14 knots	43,000	57,800	49,000	55,400	61,000	71	95	80	91	100
15 knots	50,400	66,700	56,200	62,800	69,800	72	96	81	90	100
16 knots	58,900	76,500	63,900	71,400	78,900	75	97	81	90	100
17 knots	68,800	87,300	73,900	81,100	88,900	77	98	83	91	100
18 knots	80,000	99,000	92,400	92,000	100,000	80	99	92	92	100
19 knots	94,500	112,000	....	104,200	112,000	84	100	......	93	100
20 knots	113,000	127,300	....	118,000	125,900	90	101	......	94	100
21 knots	136,500	145,200	....	133,000	141,800	96	102	......	94	100
22 knots	166,000	169,000	....	150,000	160,700	103	105	......	93	100
23 knots	203,200	199,000	....	....	184,500	110	108	......	......	100
24 knots	247,000	238,000	....	....	218,000	113	109	......	......	100
24.5 knots	....	261,800	....	....	244,000	......	107	......	......	100
25 knots	....	286,400	....	....	....	......	......	......	......	......

AUXILIARY MACHINERY.

Conclusions as to relative steam consumptions of auxiliaries, for accuracy of comparison, should be based upon close regulation of such machinery in accordance with actual requirements. The quantity of condenser cooling water required, to cite an example, is largely dependent upon sea temperature, and in consequence this becomes a controlling factor in speed of circulating pumps. Moreover, for each speed of vessel, under conditions existing at the time, there is a point of regulation which gives minimum steam expenditure for each auxiliary. On main (IV) steam-consumption tests close regulation of auxiliaries was not attempted, but on coal consumption tests every effort was made to reduce steam thus expended to a minimum. Comparisons therefore are based upon results of these trials as shown in detail in Table R.

The various auxiliary machinery installed on the three vessels may be classed arbitrarily under the following three heads: (1) Like auxiliaries whose steam consumption depends largely upon efficient condition of working parts and speed of operation, and which class includes a large percentage of the auxiliaries of the three ships. Steam expenditure for these auxiliaries should not be widely different under like conditions of use. (2) Main condenser auxiliaries: Disregarding steam used by main circulating pumps, it will be seen in Table R that

the *Birmingham's* condenser equipment required the least steam expenditure during all trials, except at full power, when the *Salem* used 6 per cent less. The greatest expenditure was on the *Chester*, indicating that the *Salem's* wet-and-dry vacuum pumps are less expensive in steam used than the *Chester's* augmenters and air pumps. It should be pointed out, furthermore, that with exception of circulating pumps, which are similar, the main condenser auxiliaries differ radically in type, which makes comparison of their steam expenditures desirable and important. (3) Forced-draft blower installations: Based on steam consumption per hour per I. H. P., the *Chester's* equipment shows a variable gain in economy, as compared with the other two vessels, of 6 per cent at low powers (5 I. H. P.), extending to 16 per cent at high (35 I. H. P.) powers.

TABLE R.—*Steam consumption of auxiliaries, in pounds, per hour.*

COMPILED FROM SUMMARY SHEETS OF COAL CONSUMPTION TESTS.

Test No.	Main air pumps.		Wet vacuum pumps.	Dry vacuum pumps.	Augmenters.	Main circulating pumps.			Steam consumption of main condenser auxiliaries (exclusive of main circulating pumps).			Steam consumption of auxiliaries necessary to propulsion.—includes main condenser auxiliaries; forced-draft blowers; oil pumps: Salem, Chester.			Steam consumption of auxiliaries not necessary to propulsion.			Total steam consumption of auxiliaries.		
	Birmingham.	Chester.	Salem.	Salem.[a]	Chester.	Birmingham.	Salem.	Chester.	(a) Birmingham.	(c+d) Salem.	(b+e) Chester.	Birmingham.	Salem.	Chester.	Birmingham.	Salem.	Chester.	(n+q) Birmingham.	(o+r) Salem.	(p+s) Chester.
	a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v
1	2,220	3,550	958	2,600	1,971	985	1,230	800	2,220	3,558	5,521	5,705	7,963	9,006	8,018	6,978	5,899	13,723	14,941	14,905
2	2,400	2,600	974	2,636	2,032	1,060	1,560	1,160	2,400	3,610	4,632	7,696	11,951	9,314	8,157	6,190	5,006	15,853	18,141	14,320
3	2,970	4,850	1,099	2,598	1,890	1,460	3,360	1,900	2,970	3,697	6,740	13,930	20,412	15,680	7,244	7,184	5,641	21,174	27,596	21,321
4	4,540	5,500	1,662	2,585	2,291	2,050	3,380	3,585	4,540	4,247	7,791	35,883	40,532	30,840	3,578	3,584	3,215	39,461	44,116	34,055

COMPARISON OF ABOVE BASED ON BIRMINGHAM (AUXILIARIES) AS 100.

1	100	160	43	132	100	100	125	81	100	160	249	100	140	158	100	87	74	100	109	109
2	100	108	41	130	100	100	147	109	100	150	193	100	155	121	100	76	61	100	114	90
3	100	163	37	137	100	100	230	130	100	124	227	100	147	113	100	99	78	100	130	101
4	100	121	37	113	100	100	165	175	100	94	172	100	113	86	100	100	90	100	112	86

[a] Comparison of dry vacuum pumps with augmenters as 100.

TABLE S.—*Coal-consumption tests.*

No. of test.	Approximate speed.	Duration of test in hours.	Birmingham.				Salem.				Chester.			
			Average pressure in fireroom in inches of water.											
			Number of boilers in use.	No. of fireroom.			Number of boilers in use.	No. of fireroom.			Number of boilers in use.	No. of fireroom.		
				1	2	3		1	2	3		1	2	3
1	10	96	3	....	....	0.694	4	....	0.325	....	4	....	....	....
2	15	50	8	....	0.546	.788	8	....	.407	0.418	7	....	....	....
3	20	98	12	1.227	1.214	1.174	12	1.57	1.58	1.554	12	....	....	[a]0.9
4	([b])	[c]24	12	5.077	4.942	5.807	12	5.08	4.84	5.11	12	3.03	3.03	3.03

TOTAL EVAPORATION PER HOUR—BLOWER STEAM—NET AVAILABLE STEAM.

			Total evaporation.	Blower steam.	Net available steam.	Total evaporation.	Blower steam.	Net available steam.	Total evaporation.	Blower steam.	Net available steam.
			A [d]	B [e]	(A-B) [f]	C [d]	D [e]	(C-D) [f]	E [d]	F [e]	(E-F) [f]
1	10	96	32,871	990	31,881	44,674	873	43,801	36,097	....	36,097
2	15	50	67,438	1,961	65,477	84,280	2,219	82,061	74,596	....	74,596
3	20	98	129,176	5,200	123,976	163,111	7,385	155,726	134,270	525	133,745
4	([b])	[c]24	271,545	18,493	253,052	293,103	24,820	268,283	314,161	6,510	307,651

PER CENT OF TOTAL EVAPORATION PER HOUR—BLOWER STEAM—NET AVAILABLE STEAM.

1	10	96	100	3	97	100	2	98	100	0	100
2	15	50	100	3	97	100	3	97	100	0	100
3	20	98	100	4	96	100	4	96	100	0.4	99.6
4	([b])	[c]24	100	7	93	100	8	92	100	2	98

[a] Forced draft used intermittently 50 hours.
[b] Maximum speed: Birmingham, 24 knots; Salem, 24.32 knots; Chester 25.08 knots.
[c] Birmingham discontinued this trial after twelve hours.
[d] Total water evaporated per hour, in pounds.
[e] Steam used per hour by forced draft blowers, in pounds.
[f] Net available steam per hour, in pounds.

PROPULSIVE EFFICIENCY.

Due to dissimilar propeller efficiencies, it is obvious that comparison based on steam consumption of the machinery (either main or inclusive of all auxiliaries) per horsepower would not indicate the relative economies of the three vessels with the same degree of accuracy as the steam used per knot. As illustrating this, results of coal-consumption trials may be cited: For example, on trial No. 3 (about 20 knots) the average horsepower (shaft) of the main engines or turbines necessary to propel each of the three vessels, as recorded in Table O, is, *Birmingham*, 7,120.5; *Salem*, 6,883.3; *Chester* (5-turbine combination), 8,374.

BOILER INSTALLATIONS.

In order to effect a true comparison covering steam-generating appliances on the scout vessels, it is essential to take into consideration not only boiler efficiencies, but, as well, expenditures of steam (forced-draft blowers) necessary in conjunction with service operation of the plants. Differences in boiler design have been pointed out, and it is to be noted that due to tortuous passage of gases of combustion across heating (tube) surfaces of the boilers of the *Birmingham* and *Salem* higher fireroom air pressures must be maintained compared with the *Chester's* boiler installation to burn the same weight of coal.

Tables C and D show boiler efficiencies under varying rates of combustion, and from an inspection of these tables it will be observed that under similar conditions as to air pressure there is a wide difference in the amount of coal consumed. It may be further pointed out, in this connection, that on 10 and 15 knot coal-endurance runs the *Chester* steamed under natural draft, while on the other two vessels with the number of boilers in operation forced draft was a necessity. It will be apparent, therefore, that comparison of steam-generating apparatus should be based upon percentage of available net output of steam, in addition to boiler efficiency, as recorded in Table S, compiled from results of coal consumption tests.

Tables (1–106) of data, machinery cuts (pls. 1–15), and performance curves (pls. 16–88), are attached to this report as an appendix.

Respectfully submitted.

F. H. Bailey,
Captain, U. S. Navy, Senior Member of Board.
Gustav Kaemmerling,
Commander, U. S. Navy, Member.
W. W. White,
Commander, U. S. Navy (Retired), Member.

The Chief of Bureau of Steam Engineering,
Navy Department, Washington, D. C.

O